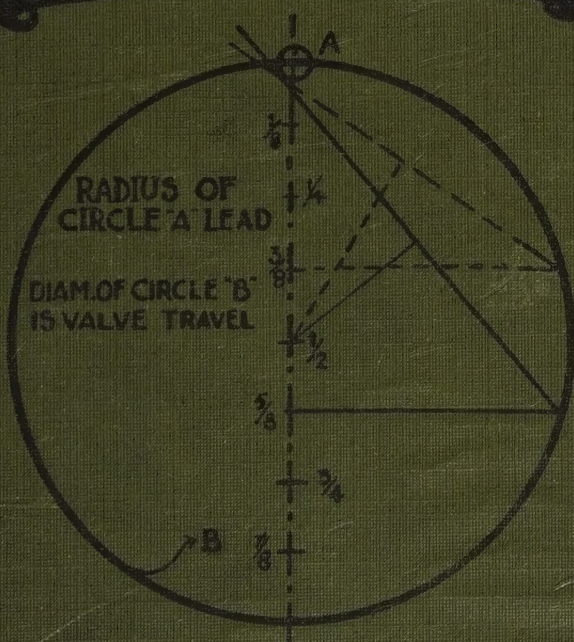


$\frac{3}{8}$ / $\frac{1}{6}$
NET

BROWN'S SLIDE VALVE FOR ENGINEERS



JAMES BROWN & SON

GLASGOW

BROWN'S
SLIDE VALVE FOR ENGINEERS

BROWN'S

Slide Valve for Engineers

Written specially for Sea-Going Engineers and
all Candidates for Second and First-Class
Board of Trade Certificates

BY

W. T. THORN

Principal, Northumberland Correspondence Academy (late W. H. Thorn & Son)

WITH NUMEROUS DIAGRAMMS



[*Entered at Stationers' Hall*]

GLASGOW
JAMES BROWN & SON, Nautical and Engineering Publishers
52 TO 58 DARNLEY STREET

1914

PREFACE

IN the belief that a work dealing with slide valves and gearing, giving all the necessary information to enable one to thoroughly understand the great importance of these parts in any reciprocating steam engine, and at the same time not to bring in a mass of more or less intricate matter which the marine engineer preparing for a Board of Trade examination does not require, has led to the Author and Publishers bringing out a work which they trust will prove of much help not only to engineers as above, but to all who have to look after engines fitted with slide valves.

Particular attention is directed to the style in which this book is written, the Author having put before his readers the information in as nearly as possible the same way as though he were speaking to them, thus the severe academic style is absent and the book should therefore appeal more to the average engineer.

Believing that the sphere of usefulness of such a work is considerably enhanced by its being well illustrated, there will be found a very large number of figures, putting the different actions, details, and arrangements in a clear way before one ; and the best thanks of the Author are due to Messrs. G. & J. Weir, Ltd., Cathcart, Glasgow, and Messrs. The Worthington Pump Co., for so kindly granting loan of blocks and placing at our disposal much useful information in regard to their justly famous pumps.

Lastly it is hoped that the size and price of this book are such as will place it well within reach of a large percentage of the thousands of engineers of to-day, and if to those who read it, it makes their knowledge of the slide valve more than they had before, then the Author will feel that his aim in writing this work has been achieved.

W. T. THORN,
Principal.

NORTHUMBERLAND CORRESPONDENCE ACADEMY,
NEWCASTLE-ON-TYNE,
1914.



Digitized by the Internet Archive
in 2025

https://archive.org/details/bwb_KV-682-545

CONTENTS

CHAPTER I

PAGE

What a valve does—Need for free exhaust—The “face” of a slide valve—Names of the different edges of valve—Elementary slide—Positions of valve as regards steam piston—Valve with steam lap—Definition of steam lap—Travel of valves—What steam lap results in—Cushioning—Lead—Engines most suited to have steam lap, etc.—Minus lap or exhaust lead—Exhaust lap—Engines suited to have minus or exhaust lap—The different laps and their effects	1-21
--	------

CHAPTER II

Driving valve correctly—“Line of throw”—Correct placing of sheave in relation to crank for an elementary slide—Steam inside and steam outside—Correct placing of sheave for valve with steam lap—“Keyway” diagrams—Iron template for transferring position of key—Angular distance between lines of throw—Movement of sheave to give a certain lineal movement to valve—Putting engine on top centre—Finding keyway position actually on shaft—Setting a slide valve—Measuring lead by means of a wedge	22-31
---	-------

CHAPTER III

Effect of liners under eccentric rod foot—Difference between changes effected by liner and by sheave—Valve with steam on inside—Various cases of valve alteration, changes in leads, etc.	32-39
---	-------

CHAPTER IV

Different types of slide valves—Double ported—Piston type with full description—“Tric” form of valve	40-48
--	-------

CHAPTER V

Generation of steam and economy due to high pressure—Economy not due to only expanding the steam—Comparison of amounts used in an engine, with high and low pressures	49-52
---	-------

CHAPTER VI

Link motion and linking up—Crossed and open rods—Length of eccentric rods and reversing link—Actions of a link motion during one revolution of engine—Sharpening cut off by linking up—Reducing valve travel by ditto—Movement of valve with link middled—Effect on leads by linking up—Effect on exhaust opening and closing by ditto	53-61
--	-------

CONTENTS

PAGE

CHAPTER VII

Effect of angularity of connecting rod on cut offs—Difference between the top and bottom cut offs—Endeavour to equalise the two cut offs—Lap finding diagram	62-69
--	-------

CHAPTER VIII

Patent valve gears—Hackworth—Marshall—Joy's—Benefit of using such gears—Effect by linking up upon the leads—Separate expansion valve of the Meyer type—Ability to alter cut offs by this valve whilst engine is running—Position of expansion valve eccentric—Loose eccentric motion	70-81
--	-------

CHAPTER IX

On the equalising of horse power between the different cylinders of a compound engine by means of linking up	82-90
--	-------

CHAPTER X

Friction of slide valves and the means adopted to lessen this—Loss by certain relief arrangements—Relief rings and balance pistons	91-96
--	-------

CHAPTER XI

Slide valves on such as Weir's and Worthington's pumps—Setting or adjustment of same—Peculiarities of these valves—Steam steering engine valves—Setting of—Reversal of steering gear—"Control" valve	97-109
--	--------

CHAPTER XII

Miscellaneous questions on such as finding point of cut off—Length of new eccentric and spindle, there being no old ones in place—How to put valve in its mid position—What to do if H.P. valve breaks—If the I.P. one breaks—If the L.P. slide breaks—Effect of a loose slide—Alternative to using relief rings only to minimise face friction, etc.	110-118
---	---------

FIGURE INDEX

		PAGE
1	Plain flat slide valve	3
2	Valve with steam lap, different edges named	4
3	Elementary valve in mid position	5
4	" " at bottom of stroke	6
5	" " back again in mid position	7
6	" " at top of its stroke	9
7	Valve with steam lap drawn down ready to admit steam at the top port	10
8	" " " just ready to open to exhaust	11
9	" " " showing top port just closed	12
10	" " " " bottom port full open to steam	13
11	" " " " top port open for "lead"	15
12	" " " " and minus lap, or exhaust lead	17
13	" with minus lap, to show late closing to exhaust, etc.	18
14	" with exhaust lap, to show early closing to exhaust, etc.	19
15	Diagram showing path of eccentric, and "line of throw"	22
16	Keyway diagram for outside steam	24
17	Iron template for transferring position of key	25
18	Keyway diagram for inside steam	26
19	Diagram showing angular distance between lines of throw	27
20	Difference between angular and lineal movement	28
21	Putting crank, etc., on top centre	29
22	Screeving reference or check marks on sheave and shaft	30
23	Skeleton diagram showing insertion of liners	32
24	Piston slide valve	34
25	Skeleton diagram showing liners, for inside steam	35
26	Double ported valve in mid position	40
27	" " " full open to steam	42
28	Piston slide valve	43
29	Diagonal bars and ports in piston valve chamber	45
30	Large sized piston valve	45
31	"Tric" valve in mid position	46
32	" " full open to steam	47
33	Old-fashioned fork ended eccentric rods,	53
34	Skeleton diagram for link motion, etc.	54
35	Diagram showing crank at cut off, with gear linked up	57
36	Diagram showing how travel is less when linked up	58
37	" " movement of link when in its mid position	59
38	Donkey crosshead and position of crank at cut off	63

FIGURE INDEX

[illegible]

Brown's Slide Valve for Engineers.

CHAPTER I

WHEN considering a set of reciprocating steam engines and boilers as fitted in any modern sea-going steamer, it is evident that a great similarity exists between them and the human system. In this latter we depend upon fuel in the form of food and air to go to the source of our power, viz., the stomach and lungs, which thereby become likened to the boiler of the steam engine, as this is the source of all the energy which the engines have to use a little later on. When our blood, through the process of digestion, etc., has assimilated all the good properties from the food and air, our heart forces or pumps this blood to and fro throughout our whole system, and is thus called upon to do a large amount of work. Further, we all know, probably in only a too painful manner, that if anything happens to interfere with the proper carrying out of the functions of the heart, either through lack of proper nourishment, disease, overstress, etc., that the whole human engine becomes deranged, and if still kept alive and working it does so in a very poor and indifferent manner, and we soon become ill.

Now, the heart in playing the important role of controller of the human system (or engine) can be likened directly to the slide valve of a reciprocating steam engine, because even supposing the boilers to be supplying ample steam, that is life to the engine, yet if the slide valve on the high pressure cylinder performs its duties in a bad manner, then not only will that particular engine be upset, but all those that come after it and depend upon the high pressure one for their steam will be adversely affected as well. The slide valve, therefore, can well be termed the "heart" of a reciprocating steam engine, and it is to give, in as clear a manner as possible, an idea as to what it has to perform, how best to do this, and if thrown out of adjustment, what can be done to rectify it, that the following pages have been written.

The term "slide" valve in its general sense means any valve whose face slides to and fro or moves in a rectilinear manner over the cylinder face. Thus, ordinary flat single ported valves, double ported, Meyer expansion valve and piston valves are all true slide valves, but the valves on internal combustion engines are not.

Turning now to the functions or duties of a slide valve as would be fitted to a marine engine, it may appear at first sight as though it were unnecessary to detail these, but the writer's experience of teaching has shown that at times one is rather apt to overlook one of its most important duties. Put briefly, any slide valve must open the ports and steam passages leading to each end of the cylinder, alternately to the entering steam, so that work may be done by the moving piston. It must next follow this up by opening alternately the same ports and passages to the exhaust, and this latter is in our opinion of even more importance to the running of the engine than is the entrance of the steam.

Let us go further into this latter part of a free exhaust. When the steam goes in to the port and passage it does so at a pressure of, say, 175 lbs. gross, and if we have to pass into that cylinder per stroke 8.64 cubic foot of steam, we shall require an opening for it, if we limit the velocity of the steam to 6000 feet per minute, of 25.72 square inches, and taking a steam port of 20 inches width, this gives $25.72 \div 20 = 1.28$ inches depth of that port opened for the admission of the steam.

Now, let us deal with the exhausting of this same steam. After cut off occurred in the cylinder, which for the case the figures just given were taken from, was 26 inches diameter and 3 feet 9 inches stroke, cut off at $\frac{5}{8}$ stroke, means the steam goes on expanding and hence dropping in pressure, but increasing in its volume until it has filled the *whole* cylinder with this reduced steam (note we neglect for this example the exhaust opening a little before the termination of the

stroke). Here we shall have $\frac{26^2 \times .7854}{144} \times 3.75$ feet = 14.14 cubic

feet of steam to get rid of from the cylinder, and whilst certainly the exhausting occupies a longer time than did the ingoing steam, thus helping to make up for the increased volume to be cleared away, yet remember the steam which is exhausting is much less energetic and able to clear out quickly, than it was on going into the cylinder, and thus a careful designer of cylinders and valves allows a larger volume in all steam ports and passages than is required by the condition of getting in the working steam satisfactorily, and by this giving of the larger area for *escape* he enables the greater volume of weaker steam to exhaust without checking the engine in its working. The above treats

this part more or less theoretically and in practice we like to have a moderate back pressure set up in order that the moving parts of the engine may be brought quietly to rest at the end of each stroke, and to set up this back pressure we arrange the valve so that it shall close or cut off the exhaust a trifle before the end of the stroke and so produce what is called cushioning or compression. This will be referred to later on when we have learnt more about a slide valve generally.

Fig. 1 shows in a perspective manner the simplest form of slide valve and brings into prominence its principal parts. Firstly we must recognise that all slide valves have two faces, and for an inverted

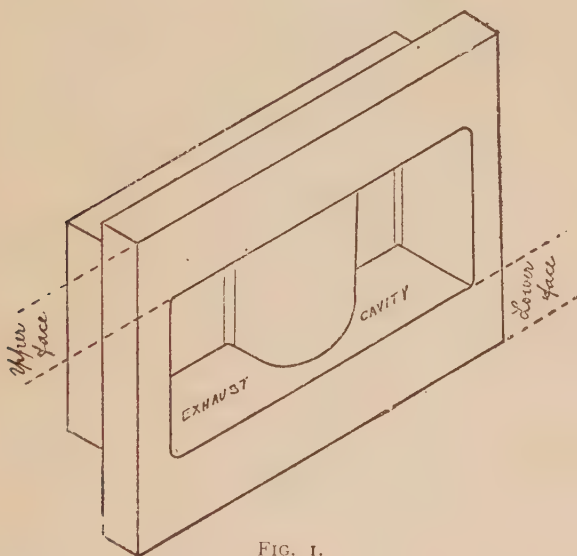


FIG. 1.

engine these are called its upper and lower face respectively and when we have this fact clearly in our mind we shall not make the common mistake of referring to a valve's face as the depth or distance from one extreme outer edge to the other. The cavity seen in the centre of the valve (fig. 1) is called the exhaust port and is formed by the upper and lower faces, together with the right and left hand side strips. A back is cast to this cavity and also down through the middle of it a tubular piece is cast and through the centre of which there passes the valve spindle, and by means of a lower collar upon this and either a collar or nuts screwed on above the valve, this latter is held in position upon that spindle. Now consider fig. 2, which shows a section through both the cylinder passages and slide valve, and gives the names of the

different edges over which the steam is admitted and exhausted. It is drawn on the assumption that the steam is admitted over the outside of the valve, which may be looked on as the original method, and whilst at the present day steam admission over the inner edges is just as common as the other, still it will be better if for the time being we do not speak of this latter style but confine ourselves to the outside steam admission. The valve is shown in its mid-position, purposely drawn away from the face on the cylinder for clearness, but of course in use the two faces are in direct contact with each other, and it is to be noted that it will then have both steam ports of the

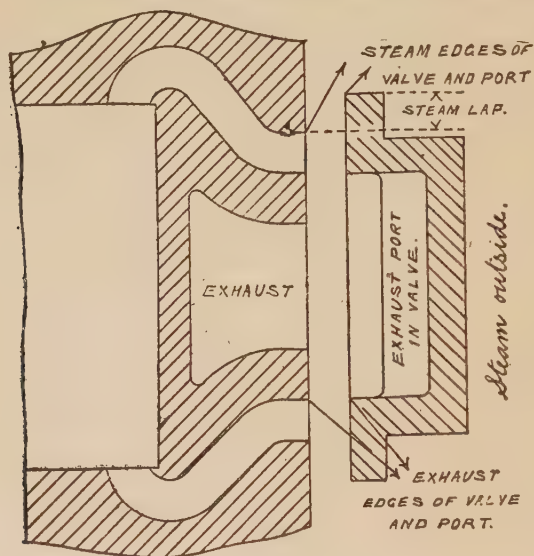
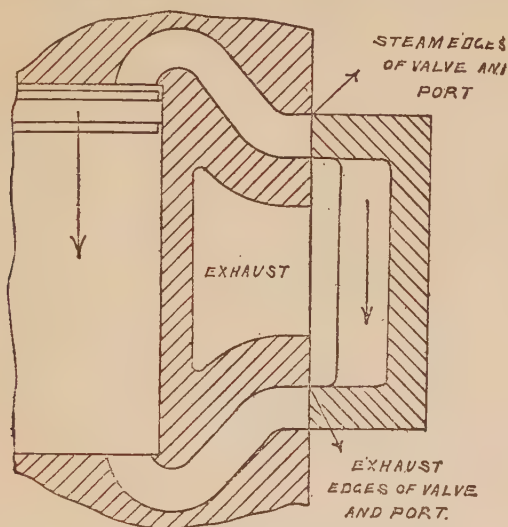


FIG. 2.

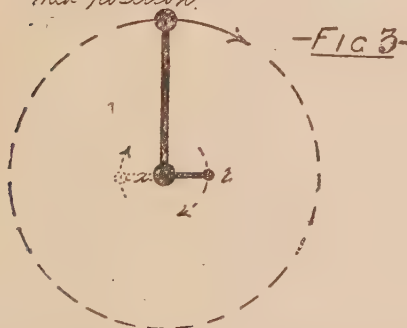
cylinder face closed to exhaust and to live steam. Further, that the outside or steam edges of the valve overlap the steam edges of the respective ports, and we thus have a valve with steam lap, and this figure should now be examined along with fig. 3, which shows a valve having no lap, and which is termed an "elementary" valve. It is seen from this figure that the depth of *each* of its faces is exactly the depth of the steam port it is covering, and this gives the particular feature of the elementary slide, because to any valve where the depth of its face equals depth of one steam port we give that distinguishing name. Whatever the kind of valve, whether piston type or flat form we can always have as a species of commencement this elementary valve, but never one which has a

depth of face at both ends *less* than the depth of ports, because if we suppose such a valve to have been fitted by accident, it will be quite evident that when it got to its mid position *both* steam ports would be then open to steam and the piston be held in equilibrium.

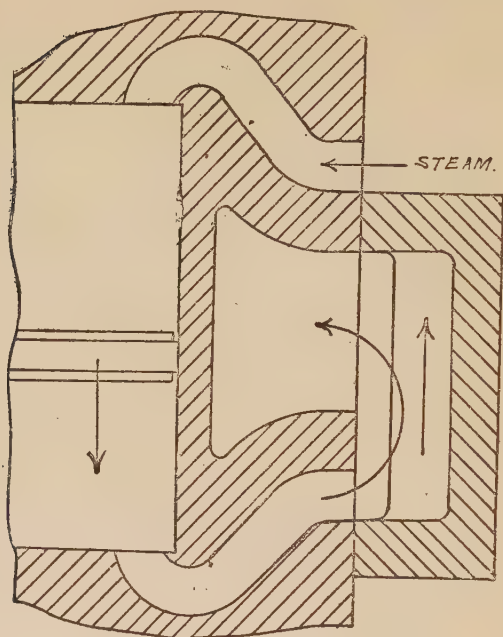
The elementary type of valve is largely used at the present day in such as steam-steering engines or in any engine that uses steam of only



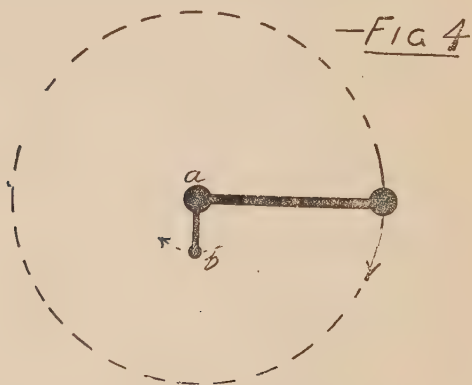
Crank on top centre, valve in mid position.



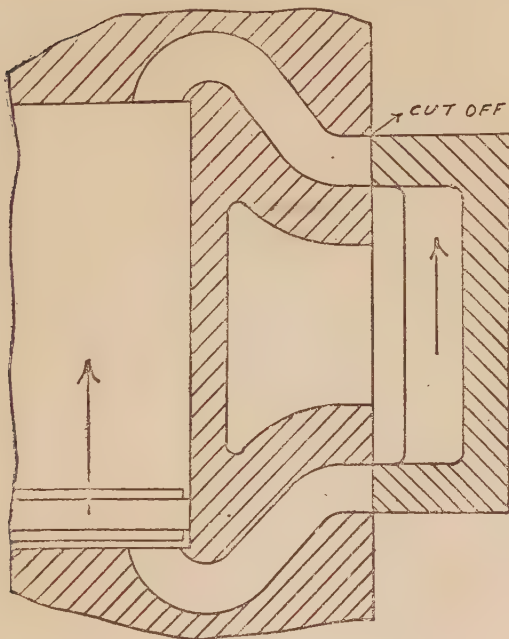
a moderate pressure, as it is really a relic of the very early days of steam engines, where the pressures were so low that in order to get a decent amount of work out of those engines, one had to design them so that steam of full pressure was carried the whole length of the stroke. This will be done with the valve as shown in fig. 3, but not



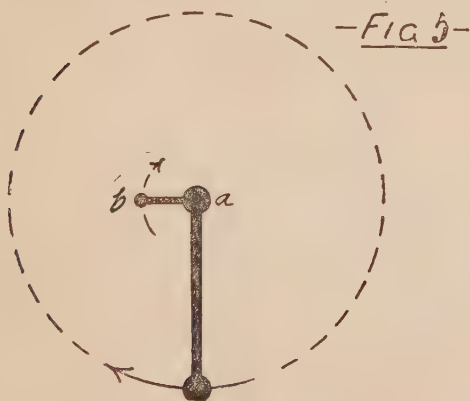
Crank in mid position, valve at bottom of its stroke.



with that of fig. 2 which, by having steam lap, cuts off the steam *before* the end of the stroke, the remainder being completed by the expansive properties, which steam of high or highish pressures possesses.



*Crank on bottom centre, valve
in mid position*



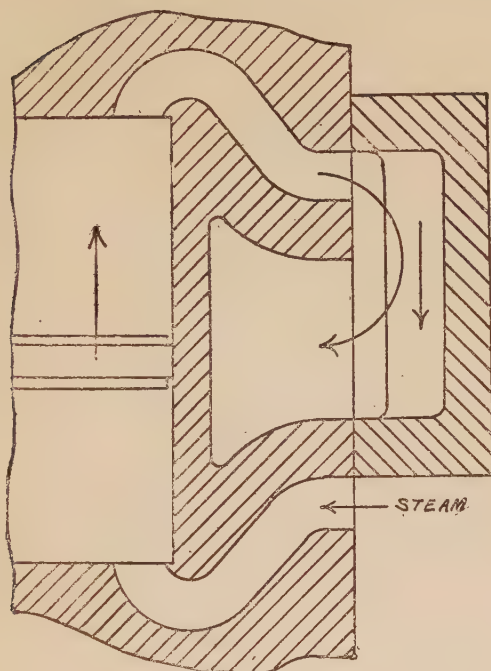
We will go more fully now into the actions of this elementary valve in regard to the position of the piston for any particular position of valve, and as we cannot take every branch of this work at once and

explain it, we must be content for the present to assume that some mechanical connection exists between, say, the shaft of the engine and the valve spindle, whereby this latter is moved at a speed proportionate to the engine in order that the valve may cover and uncover the cylinder ports properly. This part is referred to in greater detail later on in the book.

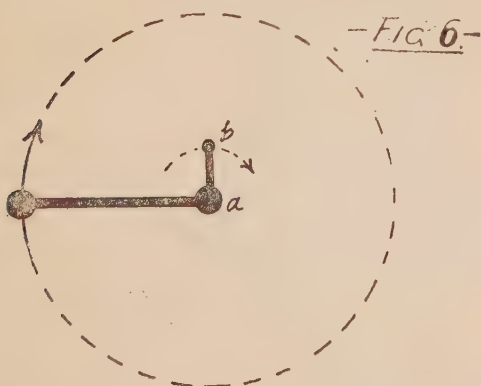
Referring to figs. 3, 4, 5, and 6, we see at first that when the crank is upon its top centre, the valve is in its mid position, and this has always to be the case with an elementary valve, because it must be so arranged that its steam edge may be on the point of opening the top port to steam in order to drive the piston downwards. Now, as the crank turns round, so does the valve operating mechanism, and when the crank has turned to the position shown in fig. 4, the piston has moved to its central position and the valve is then at the bottom of its stroke. (Note here that we neglect the effect of angularity of connecting rod.) We see, therefore, that the top port is full open to steam, and the bottom one is full open to the exhaust. Fig. 5 shows us that by the time the crank has got to the bottom centre, the valve has moved back to its mid position (going up) and has cut off the steam supply to top port, which the next instant will open to the exhaust, whilst the bottom port at the same moment will open for steam admission to drive the piston upwards. Fig. 6 shows the positions of valve and piston when the crank is again in the centre, and to sum up the actions or effects taking place with an "elementary" slide valve they are as follows:—Depth of each face exactly equal to depth of respective steam port. Steam cut off at one end, that end opening to exhaust, whilst opposite end opens for steam, all at the one instant. Cut off only occurring at the ends of the strokes, no expansion of the steam.

In the figs. 3 to 6, the smaller crank *ab* represents what is called the line of throw of the eccentric sheave which drives the valve, and which in its effect is exactly the same as a small crank.

Consider now a slide valve having steam lap, for which see fig. 2, and starting with its definition this can only be given correctly as follows:—Steam lap is the distance between the steam edge of valve and steam edge of port, when the valve is in *its mid position*. Under no circumstance must this latter part be omitted, as if so the answer is useless. Further, the definition that has just been given applies equally to whether the steam be admitted over the outer or inner edges of the valve. Whilst in this chapter we are not discussing the question of economical steam usage, and therefore do not enter into the why

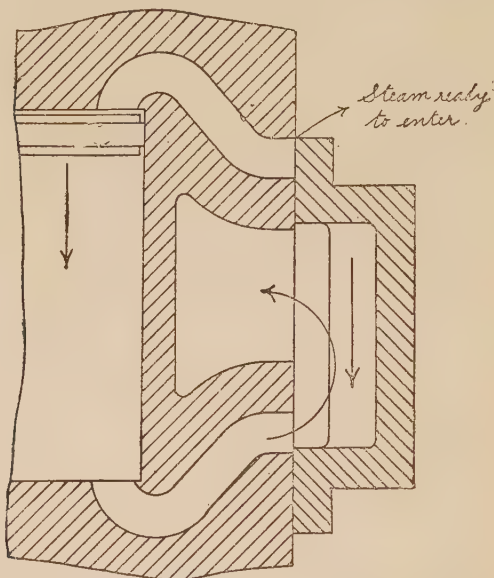


*Crank in mid position, valve
at top of its stroke.*

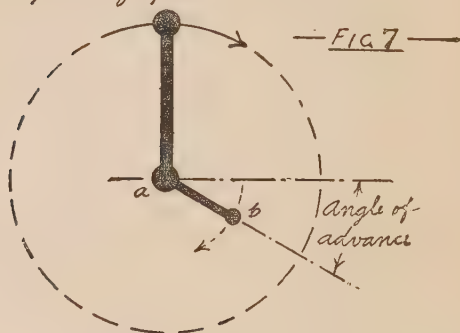


and wherefore steam lap is necessary, we confine ourselves to describing clearly the several points of great difference arising during one revolution of the crank, if the engine has a valve with steam lap. On

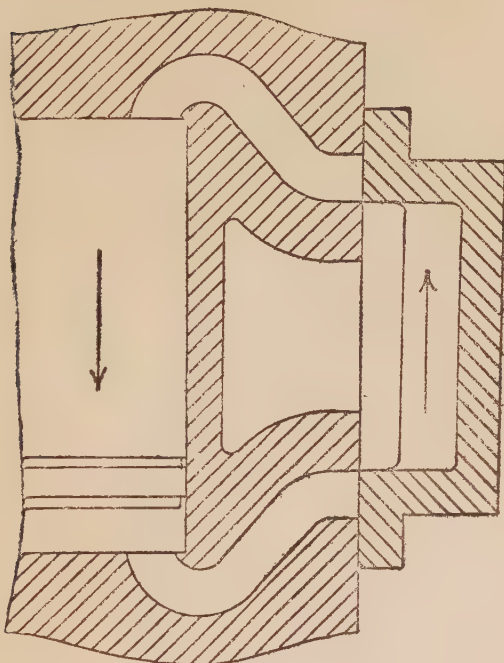
looking at fig. 7 we see at once that the length of the equivalent crank ab is much greater than that with an elementary valve, and the reason for this is as follows:—The travel of the elementary valve is equal to twice the depth of steam port opening, and if we design the slide so as



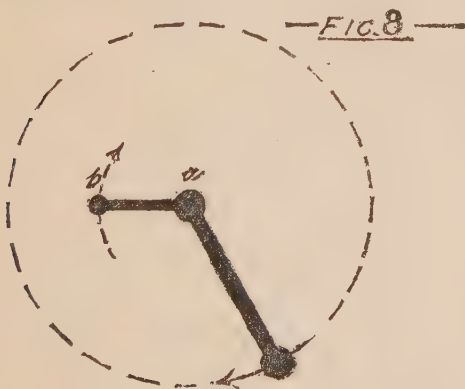
Crank on top centre, valve just opening port to steam.



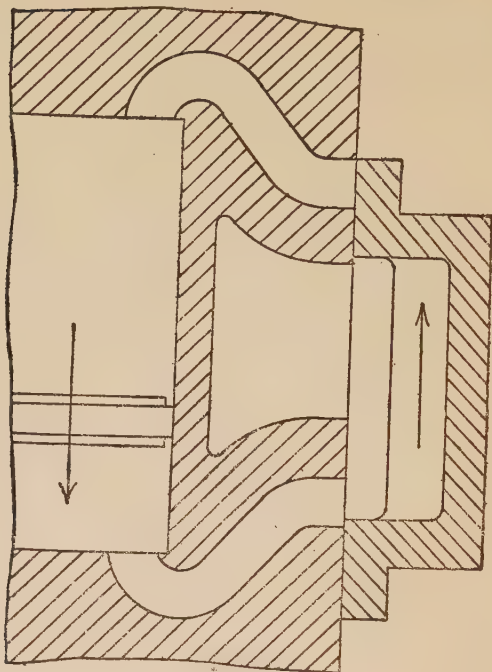
to open the whole depth of each steam port to steam admission, then that valve's travel will equal twice the steam port depth. But it is very evident from looking at fig. 7 that if we still wish to open the whole depth of steam port, we shall have first to move the valve down for an



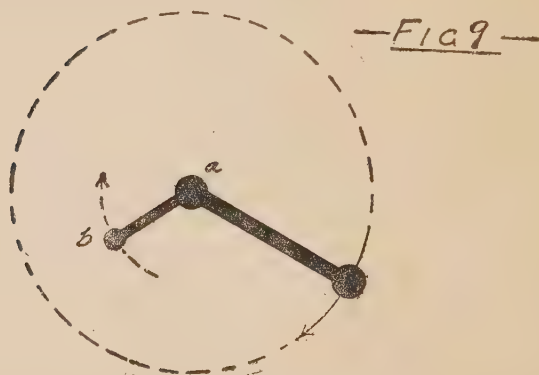
VALVE JUST OPENING TOP PORT
TO THE EXHAUST.



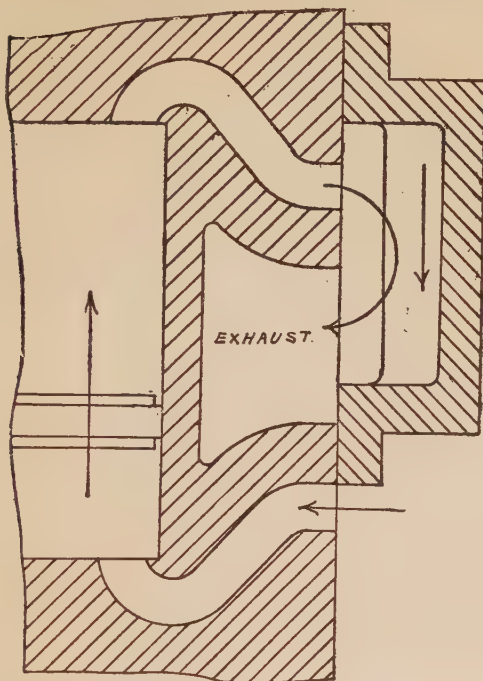
amount equal to the added steam lap, hence the total movement downwards is the lap + port opening, and for the bottom the movement upwards is also the bottom port opening + bottom lap, or the actual



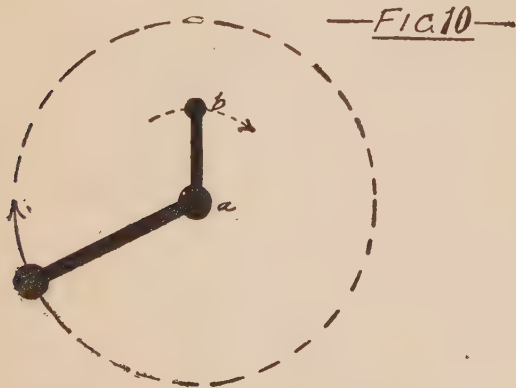
STEAM CUT OFF AT TOP PORT.



total travel of this valve is 2 (lap + port opening), if the same lap, etc., at both ends. If this should not be the case, then the travel equals top lap + top port opening + bottom port opening + bottom lap.

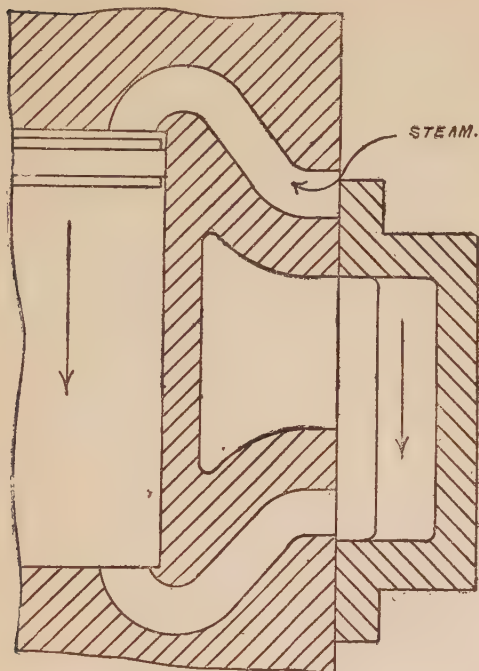


*BOTTOM PORT FULL OPEN TO
STEAM.*

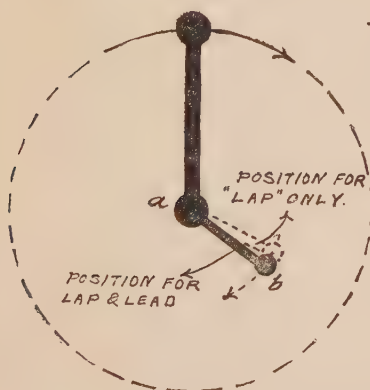


The next point to notice is that the small equivalent crank ab is placed further round the shaft by an amount termed the "angle of advance," and this is necessary in order to pull down the valve such an amount

as will make its steam edge just clipping with the steam edge of the port, when the piston is on its top centre. (Note we do not as yet speak of "lead.") Notice also that the underside of the piston is open to the exhaust then, and as the valve must move yet further down in order to admit the proper amount of steam, that the exhausting port of the cylinder remains full open for an appreciable part of the engine's stroke and thus gives greater freedom for the exhaust steam to escape. Fig. 9 shows where the piston will be when steam is cut off, as we neglect the effect of the angularity of connecting rod, this being treated specially later on in the book. On comparing this with fig. 5, it will be seen that having given the steam lap results in the cut off of the steam *before* the piston has completed the whole stroke, thus taking advantage of the fact that steam in expanding is capable of doing work. Fig. 8 shows how this top port is just opening to the exhaust and that part of the piston's stroke, which is represented by the difference in positions of figs. 8 and 9, tells us the amount that is done by "expansion" only, and whilst it does not appear to be very much from these sketches, yet by arranging for an earlier cut off, we can get a good deal of work from the expansion only. Fig. 10 shows the bottom port full open to steam and should be compared with fig. 6, and it will be seen that now, owing to having steam lap on the valve, the crank is a good deal *short* of the half stroke, as it was in fig. 6. We begin to see therefore that when steam lap has been given to a valve, everything about the steam distribution is made earlier and this is only what would be expected from looking at the lower part of fig. 7. Refer next to fig. 8 again, where it is seen that the bottom port has just closed to the exhaust, so confining the steam that did not have a chance to escape from the cylinder, with the result that the piston, still continuing its stroke, compresses this steam and so raises its pressure, thus setting up what is called "cushioning" or "compression," thus getting a higher pressure in order to bring the reciprocating masses of piston, rod, crosshead, and upper part of connecting rod quietly to rest at the termination of each stroke. We must, of course, remember that whilst we write of certain actions taking place at the one port, these same effects are produced at the other, when valve, etc., gets into their similar positions. This cushioning is the pressure we spoke of on page 3, and is entirely due to the valve having steam lap. An elementary valve could not produce it for the reason that it only shuts off the exhaust of one port at the instant that same one is ready to get its working steam, which means the piston is already at the end of its stroke. This is the



TOP PORT OPEN FOR "LEAD"



—FIG 11—

principal reason why an engine fitted with elementary valves always turns the centres with more or less of a knock, especially if she is running fast.

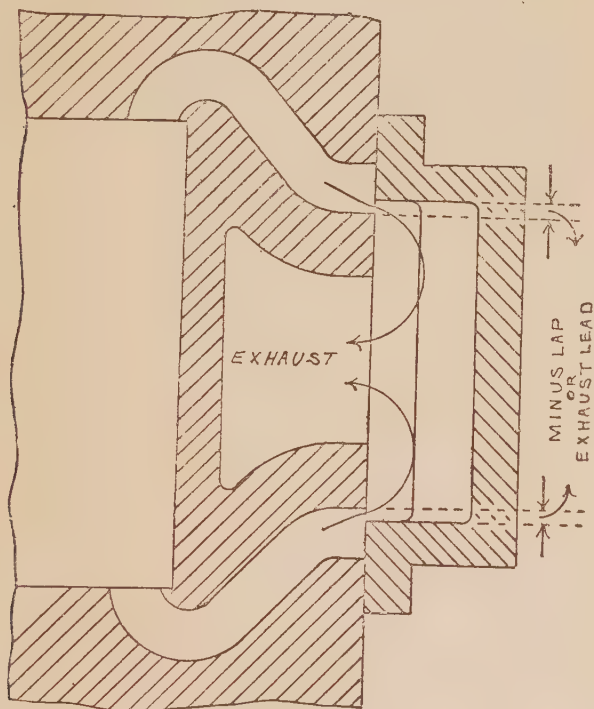
Care must be taken to see that this back pressure does not get too great, as even when moderate it means a slight loss of useful work from that cylinder, but this is more than compensated for by the smoother running of the engine. If, however, it be allowed to get too great, by, say, an oversight in the design of the valve, then the engine will, from this very high back pressure alone, experience much difficulty in turning each centre, and so its useful horsepower will be detracted from.

Having brought in the matter of a cushioning pressure, it is opportune to speak of "lead," probably one of the most important details connected with a slide valve, and one of the things we are mostly concerned with when altering or adjusting a valve's setting. In fact, we devote a special chapter of this work to "valve setting."

Fig. 11 shows the valve drawn still further down than was required merely for steam lap, and we do this in order that the port be some *little* way open to steam, when the piston is at the end of its stroke and thus the definition of "lead" is obtained. It is very evident that while the parts shown in diagram form by fig. 11 have been moving for one complete revolution of the crank, that the slide valve *started* to uncover the top port, a little *before* the piston reached its highest point (end of stroke) and hence as the live steam was entering the cylinder it met the advancing piston, and tended to push it back, but which, of course, it did not actually do. However, it is plain that a counter or back pressure was set up and so remembering that there is cushioning at both ends of the cylinder, it is clear that the lead steam on entering acts exactly the same as does the steam confined and compressed by the moving piston, viz., they both check the movement of the reciprocating masses and ensure the turning of the centres with quietude. Further by admitting the lead steam we ensure there shall be something ready to start the parts again in movement on their next stroke. At this point a caution to our readers, which is, do not give too *large* leads to an engine, as if you do you take more out of it than is gained. Remember that the amount of lead that can safely be given is governed largely by the size and speed of the engine, thus in small high speed ones, possibly $\frac{1}{32}$ inch is all they will stand, whilst with large heavy L.P. cylinders of a marine job as much as 1 inch may be required. Before we can say if a lead is too great we must know exactly the engine itself.

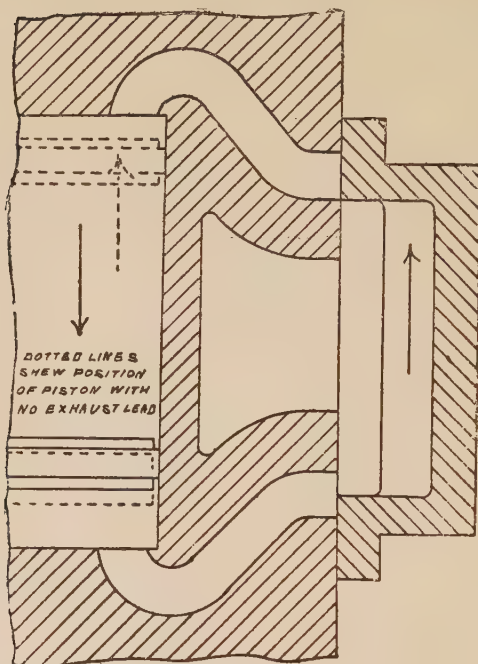
Having written at some length upon the effects of steam lap, let us state what kinds of engines are best suited to have it on their slide valves. Any engine of heavy moving parts and which makes a large number of revolutions per minute should have both steam lap and lead.

For an engine whose parts were not so heavy and with perhaps a still higher number of revolutions, then the steam lap can be retained, as it will be wanted to cut off the steam at an early part of the stroke, but the leads should be *less*, top and bottom. For an engine of very light parts and a very high speed of revolution we may have to design its valve with what is called "minus lap" or exhaust lead, in order to free the escaping steam as much as possible, also its leads will be very little, and steam lap will be retained.

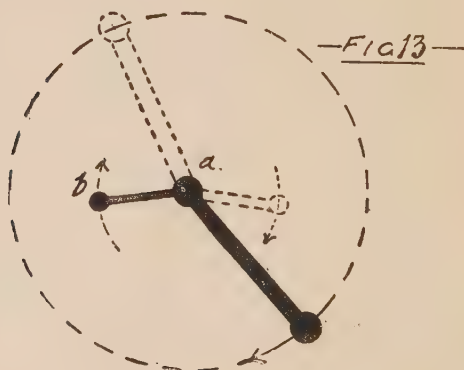


—FIG 12—

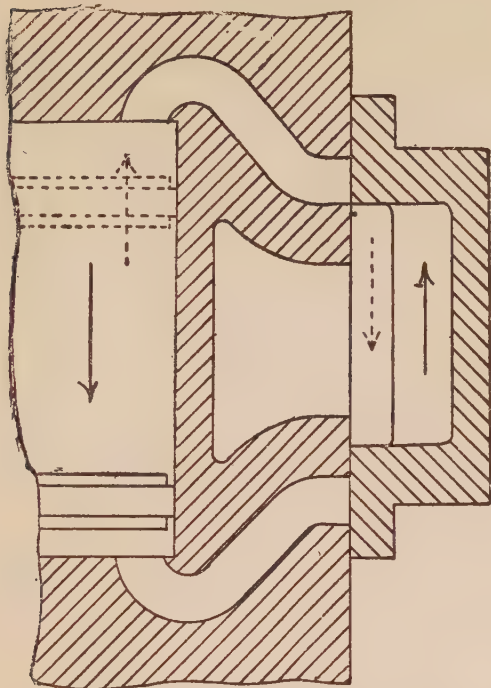
Having introduced minus lap, or what is much better called "exhaust lead," we turn to fig. 12 which shows a valve in mid position and having then both the ports of the cylinder open to the exhaust and the effect of this is that the exhaust is *kept* open longer and also closes later, but opens earlier and it is this latter which gives the name "exhaust" lead. Looking at fig. 8 let us imagine that its exhaust edges did not coincide with those of the respective steam ports, but that the depth of its exhaust port was *more* than equal to the sum of



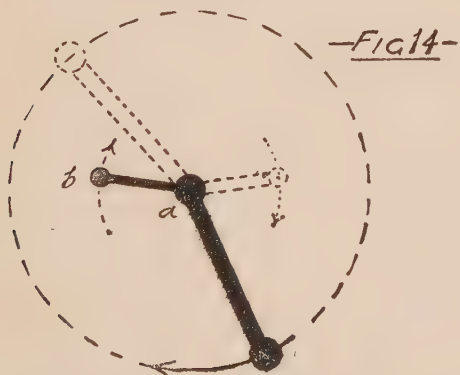
TOP PORT OPENING TO EXHAUST



the exhaust port of cylinder plus the two bars, we can then quite easily see that with the piston, crank and valve in the same position as they all occupy in fig. 8, that the top port would already be open to a certain extent and so, under the new condition of a valve with minus lap or



TOP PORT OPENING TO EXHAUST



exhaust lead, but keeping its steam edges and lap the same as before, when the steam port now starts to open to the exhaust the piston will not be so far down on its stroke, in fact, fig. 13 shows where the parts will be and should be carefully compared with fig. 8. Neither the

length of the equivalent crank *ab* nor yet its angular position in respect to the crank itself is changed in the least for a valve with this minus lap. Valves of light high speed dynamo engines are sometimes given a little of this exhaust lead in order that the piston in its very rapid travel up and down may not catch the exhaust steam and so receive a check, because not only would this prove harmful to the engine itself, but the maximum speed would never be got from it.

In the lower part of fig. 13 the dotted lines show the position taken by the crank when the top port has just closed to the exhaust, and it is to be noted that it is very near the end of the stroke, whilst the black crank indicates the whereabouts of the piston, on top port opening to exhaust and shows its position to be a good bit short of the termination of that stroke and hence we easily see how, with minus lap, the exhaust opens sooner and closes later.

Fig. 14 shows the exact opposite of the above, viz., a valve having "exhaust lap" defined as the amount the exhaust edge of valve is beyond the exhaust edge of port, when the valve is in its mid position. It opens the port to the exhaust later, closes it earlier and makes more cushioning, hence the valve of a very heavy high speed engine would have this exhaust lap. The giving of this lap does not alter the equivalent crank *ab* in any way, as this simply depends upon the lap and the steam lead.

In the lower part of fig. 14 we show in dotted lines the position of the crank, etc., as it approaches the top centre, and it shows that because we have now "exhaust lap," as the valve comes *down* to cut off the exhaust of the top port, the two exhaust edges (port and valve) meet each other sooner than before and so as the engine has to drive the valve in all cases, the crank will not turn so far when the exhaust is just closed. By the same reasoning, consider the valve rising to open the top port to exhaust, now, owing to the "exhaust lap" *more* of the face will have to travel over the port so that the two exhaust edges may again meet, and therefore the piston and, of course, the crank, must turn further round in its revolution so that the valve may be driven for this greater distance, and we see that now the exhaust opens later and closes earlier. It is to be observed that the giving of minus lap or exhaust lead to a slide valve results in its depth of face becoming less, whilst the giving of exhaust lap makes the face depth greater, this, of course, being on the assumption that the steam lap remains constant.

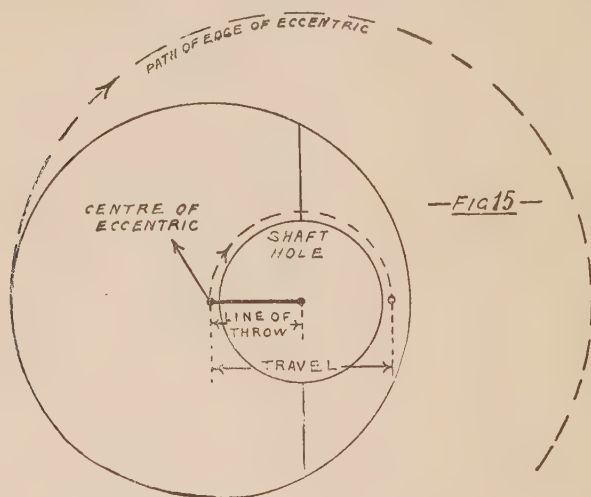
We conclude this chapter by giving a table of the different laps and the effects they produce and its careful study is advised by all readers of this book, especially those going in for their Board of Trade

examinations; and it is to be noted that in general all we have written in this chapter applies equally whether the steam be admitted over the outside or the inside edges of the slide valve. Only the actual position of the equivalent crank *ab* in regard to the main crank, becomes opposite, for the inside admission. See later chapter.

Nature of Lap.	What It Accomplishes.
Steam lap - - - {	Cuts off the steam before end of stroke; causes cushioning; allows the use of high pressure steam
Minus lap or exhaust lead - - - {	Opens port to exhaust earlier, and closes it later; gives a lead to the exhaust and reduces cushioning
Exhaust lap - - - {	Opens port later to exhaust and closes it earlier; increases cushioning

CHAPTER II

WE must now consider how the various valves, or the valves with the different laps, etc., are driven in their correct manner in relation to the position of piston. At once we see that the actual travel of valve is very much less than that of the piston and one point when designing ports and slides is to get this travel as small as possible without causing the steam to be throttled. Now, as every engineer knows there is used in the great majority of what are called valve



“gears,” one or two circular pieces of metal called, variously, eccentrics, sheaves or pulleys, but this latter word does not appeal to the writer, and so we use the term eccentric or sheave. These eccentrics get their name from the fact of their own centre being away from that of the shaft upon which they are placed (as is clearly shown in fig. 15), which also shows how their “travel” is arrived at. It should be clear to one looking at this figure that what is termed the “line of throw” is in reality the same as though a small crank had been

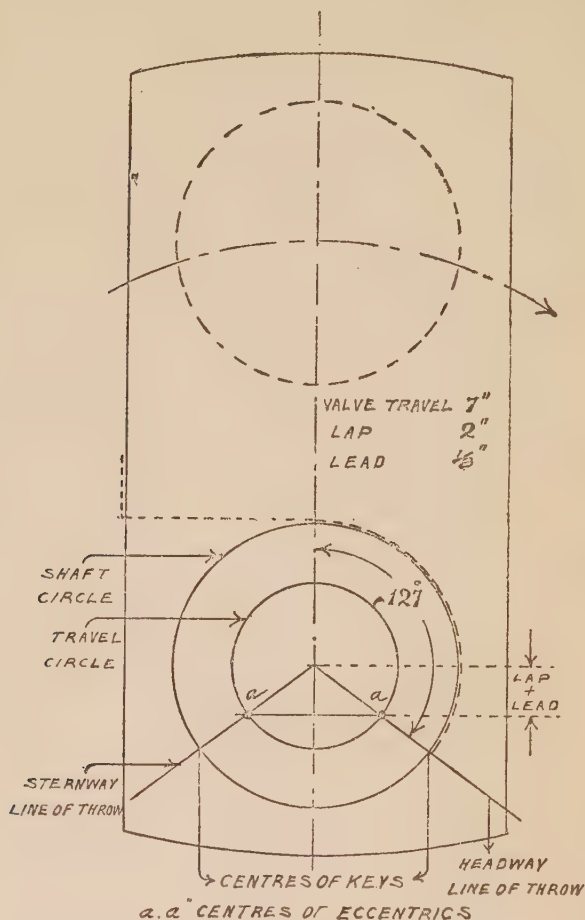
worked into the building of the shaft and whose length between centres of its pin and the shaft was the same as our "line of throw." In fact, so much is this the case that questions have been asked at examinations as to why such small cranks were not fitted, and the answer is that the eccentric makes a cheaper method of driving the valve, is more handy for any alterations in its position, works in a smoother manner and does not weaken the shaft as a little crank would do.

Whilst fig. 15 only shows diagrammatically the sheave (or eccentric) as a whole thing, it is, of course, in two parts, the larger of which for lightness is made hollow and ribbed and has a key-way cut in it, so that a key sunk in the shaft may drive this portion round with it, and then to hold that part securely on the shaft, the smaller portion of the eccentric is used and bolted to its larger neighbouring part. Sometimes one finds this larger portion made of cast iron, whilst the smaller is of cast steel in order to get greater strength. The correct placing of the eccentrics on the shaft is probably the most important thing about the erection of a set of valve gear, and it is also a factor that marine engineers are very largely concerned with, and so we devote some considerable space to it.

It is at once evident that for any given slide valve there is only one correct position, and if we suppose our valve to be of the elementary type as shown in fig. 3 we have the small or equivalent crank *ab* shown in the lower part of that figure, as representing what is called the line of throw in fig. 15, only of course to a different scale. Fig. 3 showed it was necessary to have the valve in its mid position when the crank was on the top centre, and therefore whatever is fitted like an eccentric to drive this valve must be so arranged that it also shall be at the middle of its travel, hence we must place it so as the line of throw is at 90° (a right angle) to the crank. But we are not yet sure whether this should be ahead of, or behind the crank, so we must analyse to determine this important point. Taking steam to be admitted over the outside edges of the slide, as is the case with fig. 3, and we wish the piston to move in the only direction it can, viz., downwards, it is very evident the valve must also come down in order that steam may pass into the top cylinder port, therefore we must place the eccentric leading or ahead of the crank, so that it goes down at the same time as the piston does likewise.

If, however, we assume that the live steam be supplied to the valve over its *inner* edges, then in order to get it into the top port of the cylinder, it is clear the valve will have to go *up*, and the eccentric now would need placing behind the crank, so that its line of throw may rise whilst the crank and piston come down. The dotted lines in fig. 3

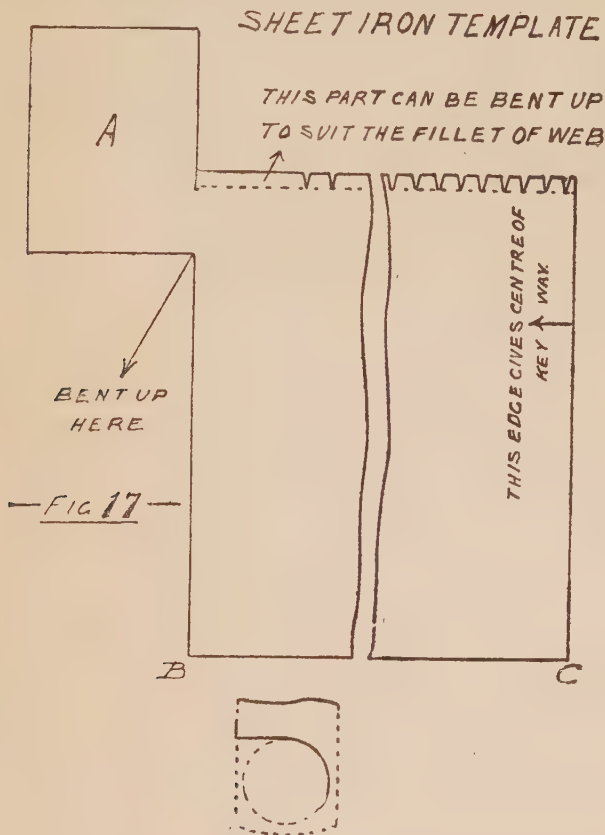
show this latter placing, and it is to be noticed that the angle is still 90 degrees. This position of 90 degrees to the crank may be called the elementary or original position, just as the valve it works is termed an "elementary" one, and it is from the fact that with the simplest form



—FIG 16—

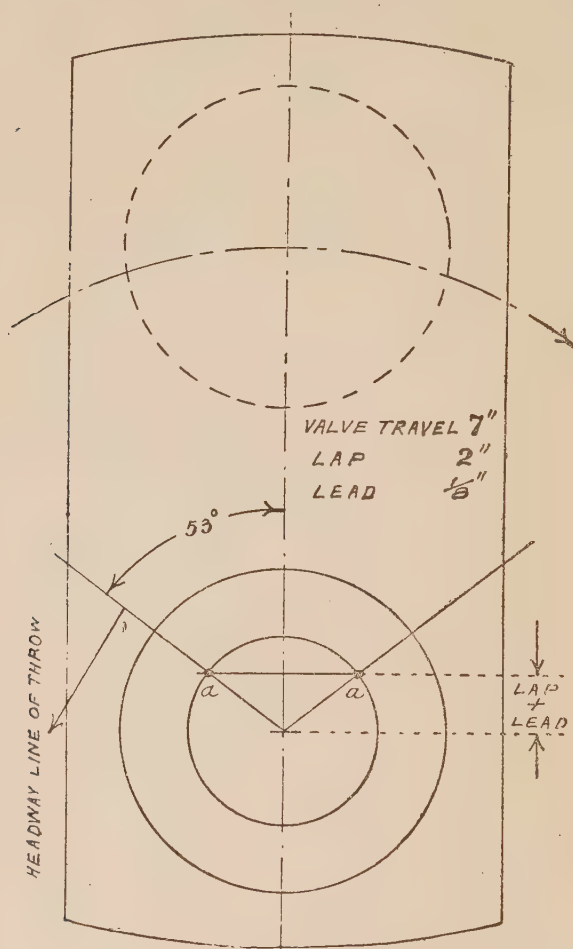
of valve its eccentric must be placed at 90 degrees, that when we have a slide with steam lap, what is called the "angle of advance" comes for the first time into being. See fig. 7 lower part. Consider next the valve as shown in fig. 7 which is one with steam lap. We have already said why the length *ab* must now be more than it was for fig. 3, also

why it is necessary to move it beyond the 90 degrees setting, and we have now to determine exactly how much it shall be advanced for a particular amount of steam lap. Taking an actual everyday case of a similar valve and one having some lead, we must construct what is called a "keyway" diagram such as is shown in fig. 16. We commence by drawing a vertical centre line and an end view of the crank



web, showing the shaft circle. Then to any given scale of equal parts, inscribe another circle whose diameter is the travel of the valve. From the centre point of the shaft, measure down on the vertical centre line an amount equal to the lap + lead, also to the same scale, and draw a short horizontal line, until it cuts the valve travel circle. Pop the two points off, so cutting, and these are the centres of the two eccentrics one for headway and one for sternway.

For the time being we only consider one, the headway, because its placing in regard to the crank is similar to that of the sternway. Next from the centre of the shaft, draw out two diagonal lines, through these two pop marks, and where each one cuts the shaft



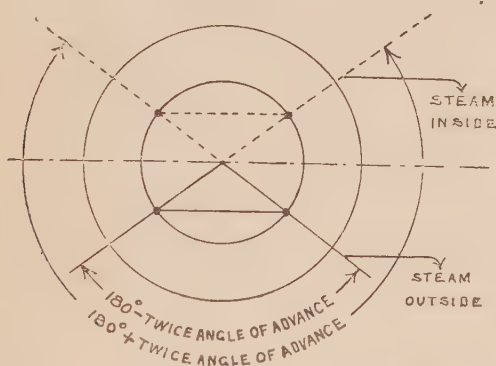
— FIG 18. —

circle gives at once the position of the respective keys or keyways on the shaft. This method is all right for finding the positions, say, on a sheet of paper, but the question arises, how is one to convey the line of throw just found, down or on to the shaft itself? The answer is as

follows :—Procure a sheet of tin or thin iron and cut out to the shape of the template shown in fig. 17, where the part marked *A* is bent up at right angles to lay against the side of the crank web as indicated in the small lower part of fig. 17. The width of *BC* must be an amount as will wrap round the shaft to a distance such that edge at *C* comes exactly to where the line of throw of the headway eccentric in fig. 16 was drawn, taking care to keep the tin pressed firmly down on to the shaft. Then scribe a line where this edge is, and that has conveyed to the shaft the position of the headway line of throw. The full breadth and length of the key can now be marked off and its bed cut in the shaft.

Fig. 18 shows us another keyway diagram only drawn now for steam admitted over the inner edges, and it will be noted that the only change consists in the lines of throw being in exactly opposite position to those

—Fig 19—



of fig. 16, which is for outside steam admission. When drawing fig. 18 be careful to measure the lap and lead *upwards* instead of downwards, and then on drawing the diagonals representing the lines of throw of each eccentric, they become at once placed opposite to that of fig. 16. When drawing such diagrams as these always place a line with arrow head to it, showing how the crank is intended to turn, for if this be omitted one cannot say which will be the headway line of throw. In both figs. 16 and 18 we have marked the angular distance between crank and eccentric, and whilst these, of course, vary for every setting of this latter, yet the figures show clearly that the line of throw, with inside steam admission, is exactly opposite that for outside admission, as $127^\circ + 53^\circ = 180^\circ$, the number in a half circle.

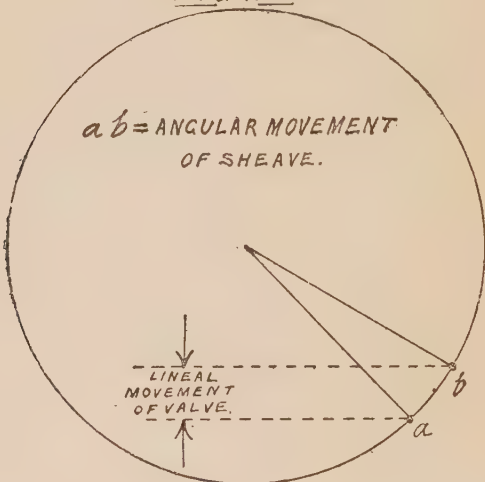
Fig. 19 is drawn to show the actual angular distance existing between the two lines of throw, in the respective cases of outside and inside

admission, and practically requires no further explanation, as fig. 7 showed us what was meant by the phrase "angle of advance."

We now group together in the form of a table the essential features of what we have been writing about and these must be memorised by all candidates for Board of Trade examinations.

Alteration Required.	What to Do.
Sharper cut off, steam inside } " " " outside }	Add steam lap, put sheave forward, or do both
Later cut off, steam outside } " " " inside }	Take off some steam lap, put sheave back, or do both
More lead top and bottom, either } with inside or outside steam } admission - - - }	Put sheave forward
Less lead top and bottom, etc.	Put sheave back

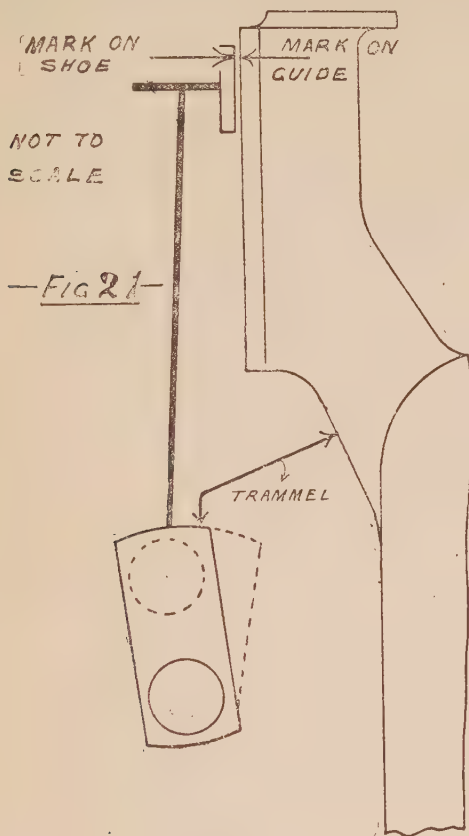
—FIG. 20—



In dealing with the alteration in slide valve setting one often reads of the phrase, put the sheave forward for $\frac{1}{16}$ inch, $\frac{1}{32}$ inch, and so on, and though it would seem to be perfectly clear what is meant by this, yet it hardly implies what at first sight one might think. To say put the sheave forward (or back) $\frac{1}{16}$ inch, really means that we must move that eccentric sheave round the shaft, either forwards or backwards, for such an amount as will give a *lineal* alteration to the slide

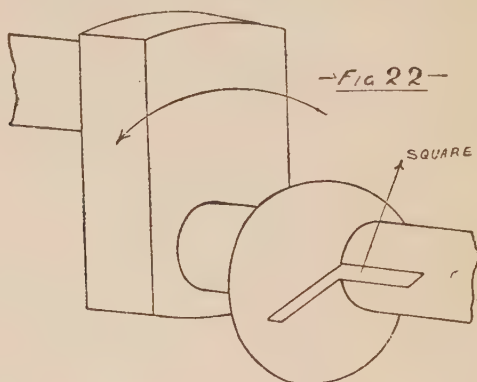
valve of the desired $\frac{1}{16}$ inch, etc. Think of what is actually happening, the sheave is being turned in a circular path, and as fig. 20 shows, a good deal greater distance must be so passed over in order that the lineal or straight line movement may be that wanted.

Before going further with the setting of a valve, let us determine how an engine's crank is put upon its top centre, because that is the position



to which reference has to be made later on. Fig. 21 shows in diagrammatic form the procedure which is as follows:—Turn the crank until it is *near* the top centre as sighted by one's eye, then mark both cross-head shoe and guide bar as shown. Make a stiff wire trammel with pointed ends to go from some pop mark made upon the condenser (or front column if one is fitted and comes handier to work from) and bent over so that it can scribe a line upon the top of crank web as indicated by the full line position in fig. 21. Now it is very evident that the

crank being a trifle short of its highest point, the crosshead also will not be quite at the top of its stroke, so as we turn the crank *past* the centre, the crosshead rises at first a little, and then comes down and when the two marks coincide again the crank will be at the same amount or angle the *other* side of the vertical. Now, therefore, scribe with the trammel in exactly the same spot on condenser or column as before, a second mark on the top of the web and with, say, a pair of compasses or a foot rule bisect the distance between these two scribe marks and reverse the turning gear so as to bring the crank back until this bisection line comes exactly under the bent over pointed end of the trammel. Doubtless this seems a rather long method, but it is accurate and that is the first condition, and also it is the style of answering favoured at the Board of Trade examination. We are now better able to present clearly the operation of setting a slide valve and will for



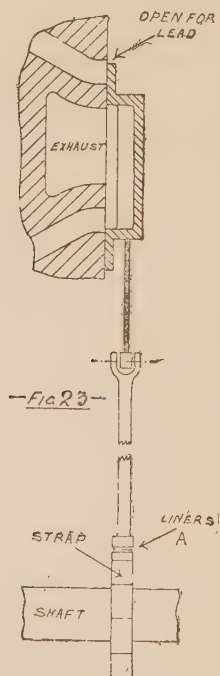
the time confine ourselves to a flat valve and one having steam over the outer edges. Piston valves with inside steam will be treated separately and later on, as at present we want to push home the easiest methods of getting principles well grasped.

Suppose we have a new crank shaft and the keyways have not been cut in it, and so it becomes necessary for us to find out where we must cut them. It is, of course, quite true that the keyway diagrams already described would give us their exact location, but occasions arise when it is requisite that this finding of position be done actually from the engine, hence proceed thus. Place the crank, piston, etc., on top centre, then put the sheave upon the shaft and just nip its bolts up sufficient to prevent play, but not enough to stop the eccentric being moved round back or forward as wanted. Then couple up its strap

and place the eccentric rods, reversing link, etc., all up, and observe that the headway eccentric rod is brought under the valve spindle, *i.e.*, the valve motion arranged in what is termed "full gear." Then having the steam chest door or cover off, so that the top of the valve can be seen, move the sheave round the shaft in such a direction as will bring the valve to open the top port to the desired amount of "lead." Then with a square, scribe lines against the eccentric sheave and along the shaft (see fig. 22), and after this tighten up the bolts of the eccentric and nip down the set pin or pins it is fitted with. Now turn the engine to the bottom centre and carefully note what the lead is thereat, and to ascertain which is at times troublesome, but a thinnish wedge may be useful for pushing into the opening of the port and the edges of this port and valve will mark the wedge so that its thickness can be measured. Next inspect the two lines scribed with the aid of the square, and see that the one on the eccentric exactly coincides with the one on the shaft, thus showing whether the sheave has moved or not, relatively to the shaft, while the engine and gear have turned through the half revolution. If the marks agree and the lead, as measured by, say, the wedge method, is correct, then the position of the keyway in the eccentric gives at once the spot where the key must be placed in the shaft.

CHAPTER III

IN this we introduce the effect upon the setting of a valve by varying the thickness of the liner or liners which are placed between the eccentric rod foot and the top flat part of the strap. Again for the time we confine ourselves to a valve taking its steam over the outside edges. Consider fig. 23, which is a more or less diagrammatic repre-



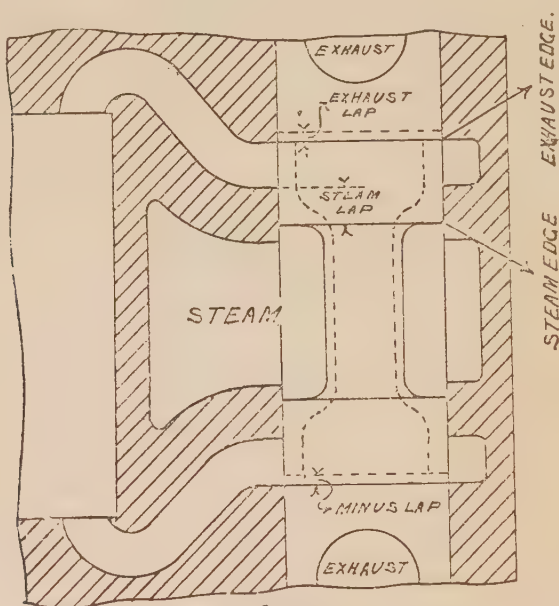
sentation of the various parts forming a valve motion (non-reversing), taken for simplicity thus, but whatever we learn of altering valve settings from this of course applies equally to a reversing engine.

The figure shows the top port open for a certain amount of lead, and

by what we have said in the previous chapter, it should be evident that there will be the same amount of lead at the bottom port, as the sheave governs both ends equally. But suppose we wished to have a slight increase of the bottom lead, and at the same time a corresponding decrease in the amount of the top lead, then for the arrangement shown in fig. 23 we could, by means of increasing the thickness of the liners indicated at *A*, lift up the whole of the eccentric rod, valve spindle, and of course the slide valve as well. It is obvious, therefore, that the amount the top port is open for will be reduced by exactly the thickness of the additional liner. A moment's thought will show that the bottom port will have its amount of opening increased by the same extent. Because, think what has happened when the liner was put in. It made the distance between the bottom edge of valve and the top of strap more than at first, just exactly as it made the distance to the top edge of valve more; hence when the engine and its gear have turned to the bottom centre, and the valve then has the bottom port open for lead, it will by reason of this increased distance have the valve further *up* than before, and so the bottom lead is made greater. Next suppose we wish to make the top lead a trifle more, while the bottom one is to be reduced by an equal amount, then we must take a liner out for the extra lead desired, which will lower the valve and its gear, and so effect what is wanted. Note that all this is true only for steam admitted over the outside of the valve, and inside steam admission will be treated on shortly. We see, therefore, that any alteration in the setting of a valve, brought about by means of a liner, yields dissimilar results at the two ends of each stroke, while alterations produced by a change in the sheave's position gives the same effect at both ends. The following table puts matters concisely, and should be carefully studied:—

STEAM OUTSIDE.					
SHEAVE ALTERATION.			LINER ALTERATION.		
How moved.	Top Port.	Bottom Port.	In or Out.	Top Port.	Bottom Port.
Forward	Lead more— Exhaust opens and closes sooner. Cut off sooner	Same as Top	In	Lead less— Exhaust opens sooner and closes later. Cut off sooner	Lead more— Exhaust opens later and closes sooner. Cut off later
Back	Lead less— Exhaust opens and closes later Cut off later	Same as Top	Out	Lead more— Exhaust opens later and closes sooner. Cut off later	Lead less— Exhaust opens sooner and closes later. Cut off sooner

It will be seen from this table that both the exhaust and the cut off are also affected by these "liner" changes, and it is clear that we cannot change one thing without the corresponding alteration of whatever other action there may be to occur at the same end. Thus, if we take the case of a liner coming out, so lowering the valve and giving more lead at top, we of course lower the exhaust edge of the valve by the same amount, and then as it moves *up* to uncover the top port to the exhaust, it does so later than before. Similarly as the valve comes down to close the port to exhaust, by virtue of its being lowered, it closes earlier.



—FIG 24—

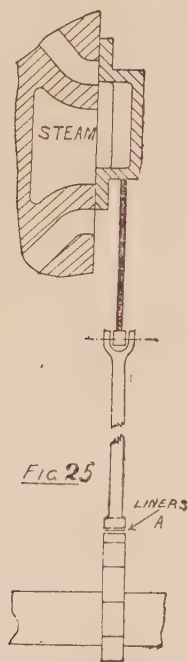
It is now time we definitely considered slide valves having the steam on their inside, and so admitting it to the cylinder ports, etc., over the inner edges. Whilst we start the explanations straight off with a valve having steam lap, yet we may state that the elementary valve will follow just the same laws or rules as the one with lap, and, in fact, with a steam steering gear it is this elementary form which are the engine's main slides, and by their getting the working steam over either the outer or inner edges, just as the "control" valve is moved, so the engine is enabled to reverse.

Fig. 24 shows a simple form of piston valve, as that is usually the

type which has inside steam, and for a description of which valve we must refer readers to the chapter on different patterns of valves.

Practically the only thing to notice is that owing to the inside admission all actions are opposite to those for outside steam, but remember there are many piston types of slide valves having their steam over the outer edges, so let no one get the idea that the piston valve can only have inside steam.

What is meant by opposite actions is as follows:—On comparing fig. 24 with fig. 7, we see with the former the valve must go *up* to



allow steam to pass to the top port, whilst with fig. 7 the valve went down. It also rose to open top port to the exhaust, whereas from fig. 24 it comes *down* to do the same thing, and we do not think more time need be spent upon this point, because a careful study of the respective keyway diagrams will make matters much clearer.

Returning then to the matter of altering the setting of slides, allowing for these two cases of outside and inside steam, we have the following:—To obtain more lead top and bottom put the sheave forward upon the shaft, *i.e.*, move it round a little in the direction the crank is turning, and to make the leads less, top and bottom, turn it

back a trifle. These remarks apply with equal truth for both cases of steam admission. To get more lead at top and less at bottom with outside steam we take a liner *out*, but with inside admission the liner is put *in* under the foot of eccentric rod. And now fig. 25 should be studied, because even though it is drawn for a flat valve, yet having steam on the inside it shows clearly that if more lead is required on the top port, and less at the bottom, that a liner must go in under the eccentric rod foot.

To produce more lead at bottom and less at the top, with outside steam, put *in* a liner, but for inside steam take it *out* for the same amount.

We have now arranged matters such that we can go finally into the question of altering or adjusting valve settings, taking due account of changes by means of the sheave only, by a liner only, by both these, and also for outside and inside steam admission.

Starting on a simple case, suppose we wish to have $\frac{1}{16}$ inch more lead top and bottom, steam outside, what are we to do? *Answer*—Put the sheave *forward* for such an angular amount as will give a lineal or straight line movement to the valve of $\frac{1}{16}$ inch, and remember if the steam should be admitted over the inner edges, it makes no difference; the sheave, even though it will now be behind the crank, will still be advanced the same amount in the direction the crank travels in.

Next suppose we want to decrease our leads by say $\frac{1}{32}$ inch top and bottom, steam outside of valve, what to do? *Answer*—Put the sheave *back* for what is termed $\frac{1}{32}$ inch (see our remarks on page 28 regarding this phraseology), and again it makes no difference whether the steam be over the outer or inner edges of the slide valve.

Now suppose we want $\frac{1}{32}$ inch more lead at top and $\frac{1}{32}$ inch less at bottom, what to do? *Answer*—Take a liner *out* for $\frac{1}{32}$ inch, if we suppose outside steam admission; but put it *in* for $\frac{1}{32}$ inch if the steam be inside, and note that now, as we have already explained, the opposite action has to take place for this inside admission to what was required for the outside.

If now we want say $\frac{1}{16}$ inch less lead at top and $\frac{1}{16}$ inch more at the bottom, what are we to do if the steam be admitted on the *inside* of valve? *Answer*—Take $\frac{1}{16}$ inch liner *out*, and so drop the slide for that amount.

Next suppose us engaged in setting a valve with outside steam in the same manner as that indicated on page 30, and we set her all correct at the top port, and with say $\frac{3}{32}$ inch lead. We then turn the engines to the bottom centre, and, either by inspection or by the wedge, find

there is $\frac{3}{8}$ inch lead, and we want only $\frac{3}{16}$ inch, what is to be done? It will be noticed that this is a different case to the simple ones first given, and will pay for a little investigation. It is, of course, obvious that the top already being set correctly must not be finally upset or altered in any way, and so whatever is done to put bottom right must leave the top as it is. Now the lead being too great at the bottom port by an amount equal to $\frac{3}{8}'' - \frac{3}{16}'' = \frac{3}{16}''$, it is necessary to put the sheave back for *half* the amount of error, so here it will be $\frac{3}{32}$ inch. But this takes off $\frac{3}{32}$ inch of the top lead as well as $\frac{3}{32}$ inch of the bottom, and we so far have not this latter lead correct, and have upset the top, so evidently there is something else to do, which is actually the case, and we next turn to the liner under eccentric rod foot. The top lead has been destroyed altogether, seeing that it was $\frac{3}{32}$ inch to start with, and we have backed the eccentric sheave for $\frac{3}{32}$ inch, therefore take out a liner of thickness $\frac{3}{32}$ inch, and so drop the valve for this amount, and restore the top lead correct, and take off the excess or remaining $\frac{3}{32}$ inch bottom lead. Put briefly, the changes here have been thus—eccentric back $\frac{3}{32}$ inch, destroys $\frac{3}{32}$ inch lead, top and bottom, liner *out* for $\frac{3}{32}$ inch destroys the other $\frac{3}{32}$ inch bottom lead and restores the top $\frac{3}{32}$ inch, so we have finally destroyed at the bottom $\frac{3}{32}'' + \frac{3}{32}'' = \frac{3}{16}''$, and leave ourselves with what we wanted, and the top lead is finally left unaltered. A rule framed from the foregoing is thus—Note the total error at either end and halve that amount, one-half cure by means of sheave movement, and the other half cure by the liner, and, of course, one must be governed by their own knowledge as to whether the sheave goes forward or back, and whether the liner has to go in or be taken out.

Next suppose a case something similar to the previous one, only having $\frac{1}{8}$ inch lap at the top instead of $\frac{1}{16}$ inch lead, and at the bottom $\frac{3}{8}$ inch when only $\frac{3}{16}$ inch lead is needed, what are we to do? You may consider the steam on the inside of the valve.

Notice here that the amount of error at the top port is exactly the same as that at the bottom, and so by putting *in* a liner for thickness of $\frac{1}{8}'' + \frac{1}{16}'' = \frac{3}{16}''$ we shall lift up the valve and so destroy the lap at top port, whilst the bottom lead will be reduced by $\frac{3}{8}'' - \frac{3}{16}'' = \frac{3}{16}''$, which is what we want.

Now consider a still more involved case. Suppose we have a lead of $\frac{5}{16}$ inch on top and $\frac{5}{16}$ inch on bottom, and we want $\frac{1}{8}$ inch top and $\frac{1}{16}$ inch bottom. Here notice that not only do we alter the leads top and bottom, but we also alter the respective totals before and after the change. The total we have is $\frac{5}{16}'' + \frac{5}{16}'' = \frac{5}{8}''$, and the total of what we

want is $\frac{2}{16}'' + \frac{3}{16}'' = \frac{5}{16}''$. The difference between these is $\frac{8}{16}'' - \frac{5}{16}'' = \frac{3}{16}''$, and the rule is then to *halve* this difference and alter the sheave for that amount, next putting one end correct by means of a liner. When this is done the other end will also be correct. Analyse this statement as follows:—Half the difference is $\frac{3}{32}$ inch, and as we do not want so much lead, put the sheave *back* for $\frac{3}{32}$ inch. This will give at the top $\frac{3}{16}'' - \frac{3}{32}'' = \frac{3}{32}''$ lead, and at the bottom it will produce $\frac{5}{16}'' - \frac{3}{32}'' = \frac{7}{32}''$. Next make, say, the top correct by means of a liner of a thickness equal to $\frac{3}{8}'' - \frac{3}{32}'' = \frac{1}{2}''$ taken out from under eccentric rod foot if steam outside, and by so dropping the valve for this $\frac{1}{2}$ inch we obtain finally $\frac{1}{8}$ inch lead at top. For the bottom there is $\frac{7}{32}'' - \frac{1}{2}'' = \frac{6}{32}''$ or $\frac{3}{16}''$, again being as we wished.

Another case of the same kind. At present we have $\frac{1}{32}$ inch lead at top and $\frac{1}{32}$ inch lead at bottom, and we wish to have $\frac{1}{16}$ inch lead on top and $\frac{1}{8}$ inch lead at bottom. Assume steam to be inside of valve.

The total lead we have is $\frac{1}{32}'' + \frac{1}{32}'' = \frac{2}{32}''$. Total lead wanted is $\frac{1}{16}'' + \frac{2}{16}'' = \frac{3}{16}''$ or $\frac{6}{32}''$, the difference being $\frac{6}{32}'' - \frac{2}{32}'' = \frac{4}{32}''$. Half this is $\frac{2}{32}$ inch, and as lead is wanted we put eccentric sheave forward to give that amount. We then have at the top a lead of $\frac{1}{32}'' + \frac{2}{32}'' = \frac{3}{32}''$ and at the bottom a lead of the same amount, neither of which are as we want them. Then, as the steam is on the inside of the valve, and we want the *bottom* port to have $\frac{1}{8}$ inch lead, we must take *out* a liner of a thickness equal to $\frac{1}{8}'' - \frac{3}{32}'' = \frac{1}{32}''$, which puts the bottom right and at the same time makes the top lead *less* by $\frac{1}{32}$ inch, leaving us with $\frac{3}{32}'' - \frac{1}{32}'' = \frac{2}{32}''$ or $\frac{1}{16}''$, which is just exactly what was needed.

Our last case with valve adjustment is probably one of the most difficult an engineer is called upon to tackle. At present say we have $\frac{1}{8}$ inch *lap* on the top and at the bottom $\frac{1}{16}$ inch lead, required $\frac{1}{8}$ inch lead on the top and $\frac{1}{4}$ inch lead at the bottom. If we treat the lap as though it were a negative lead things will work out all right. Thus the total of what we have is $-\frac{2}{16}'' + \frac{1}{16}''$ or $-\frac{1}{16}''$, and the total of what is wanted will be $\frac{2}{16}'' + \frac{4}{16}'' = \frac{6}{16}''$. The difference between these is $\frac{6}{16}'' - (-\frac{1}{16}'')$ or $\frac{6}{16}'' + \frac{1}{16}'' = \frac{7}{16}''$. Half of this is $\frac{7}{32}$ inch, and so as lead is wanted we put the eccentric forward for such an amount as will give $\frac{7}{32}$ inch movement of the valve, and we then have results as follows. At the top there is $-\frac{2}{16}$ inch, or what is better expressed as $-\frac{4}{32}'' + \frac{7}{32}''$ or $\frac{3}{32}''$ *lead*, and on the bottom we have $\frac{2}{32}'' + \frac{7}{32}'' = \frac{9}{32}''$ lead, but yet neither are correct, the top being short of what is needed by $\frac{1}{32}$ inch, hence take out a liner of this thickness, and by so dropping the valve (steam being taken to be on outside) we shall obtain $\frac{3}{32}'' + \frac{1}{32}'' = \frac{4}{32}''$ or $\frac{1}{8}''$

lead, and for the bottom we shall get $\frac{9}{32}'' - \frac{1}{32}'' = \frac{8}{32}''$ or $\frac{1}{4}''$ lead, and again both are just as we wished.

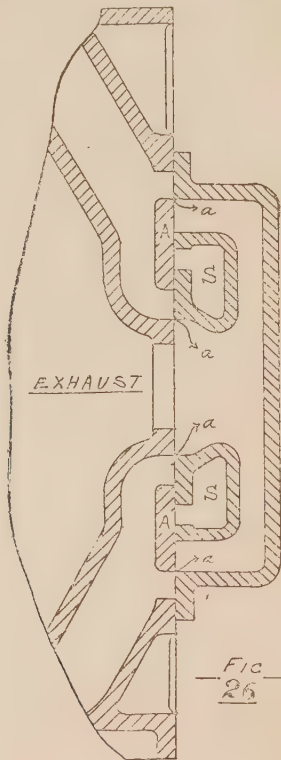
Let us now see what would have been necessary suppose this last case had had its steam *inside*. Up to the point of alteration by the liner every thing remains the same, but now as the top is short by $\frac{1}{32}''$ inch we must put *in* a liner of that thickness, and so obtain the required $\frac{4}{32}''$ inch lead on top. For the bottom we have already $\frac{9}{32}''$ inch which is too much, and so the putting *in* of the $\frac{1}{32}''$ inch liner will take off the extra lead and leave us the $\frac{8}{32}''$ needed.

It is hoped the different cases given in this chapter will prove of benefit, as it comprises about every one that will arise in everyday work, and the appended table is commended to the careful study of all engineers, whether marine or land.

LEAD WANTED.		LEAD NOW.		ALTERATION REQUIRED.
Bottom.	Top.	Bottom.	Top.	
$\frac{5}{16}''$	$\frac{5}{16}''$	$\frac{1}{4}''$	$\frac{3}{16}''$	Eccentric forward for $\frac{1}{16}''$
$\frac{1}{4}''$	$\frac{1}{8}''$	$\frac{3}{8}''$	$\frac{1}{4}''$	Eccentric back for $\frac{1}{8}''$
$\frac{1}{16}''$	$\frac{1}{16}''$	$\frac{1}{8}''$ lap	$\frac{3}{8}''$	If steam outside of valve, put a liner in for $\frac{5}{16}''$
$\frac{3}{16}''$	$\frac{1}{16}''$	$\frac{1}{8}''$ lap	$\frac{3}{8}''$	If steam be inside of valve, take a liner out for $\frac{1}{16}''$
Correct	$\frac{5}{16}''$	Correct	$\frac{3}{16}''$	Put sheave forward for $\frac{3}{16}''$ and with outside steam, liner out for $\frac{1}{32}''$
$\frac{1}{4}''$	Correct	$\frac{1}{4}''$ lap	Correct	Put sheave forward for $\frac{1}{4}''$, and if inside steam, liner out for $\frac{1}{4}''$
$\frac{3}{16}''$	$\frac{1}{16}''$	$\frac{3}{16}''$	$\frac{1}{16}''$ lap	Put sheave forward for $\frac{1}{16}''$, and if outside steam, liner out for $\frac{1}{16}''$
$\frac{1}{8}''$	$\frac{1}{8}''$	$\frac{1}{4}''$	$\frac{1}{8}''$	Put sheave back for $\frac{1}{16}''$ and if steam outside, liner out for $\frac{1}{16}''$
$\frac{3}{16}''$	$\frac{1}{16}''$	$\frac{3}{16}''$	$\frac{1}{8}''$	$3 + 1 = \frac{4}{8}$ or $\frac{1}{2}$; $3 + 1 = \frac{4}{8}$; $8 - 4 = \frac{4}{8}$ diff. in leads. Eccentric back for $\frac{1}{16}''$ giving $\frac{1}{4}''$ lead at bottom and 0 at top. Next a liner out for $\frac{1}{16}''$
$\frac{1}{16}''$	$\frac{1}{16}''$	$\frac{1}{8}''$	$\frac{3}{16}''$	Diff. in total of leads again $\frac{1}{16}''$. Eccentric back for $\frac{1}{32}''$ giving $\frac{1}{4}''$ on top and 0 at bottom. Next a liner in for $\frac{1}{16}''$
$\frac{3}{4}''$	$\frac{3}{8}''$	$\frac{1}{2}''$	$\frac{3}{16}''$	$18 - 11 = 7$, or $\frac{7}{16}''$ diff. in totals of leads. Eccentric forward for $\frac{7}{16}''$ and if steam inside, take a liner out for $\frac{1}{32}''$

CHAPTER IV

IN this chapter we are going to describe slide valves themselves in greater detail and in some of the various forms in which they have been designed. At the outset we must remember that in practically



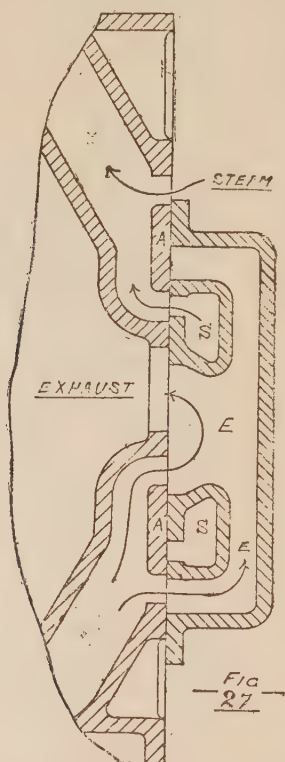
every case where the valve differs in design to what may be called the good old-fashioned flat type, the alteration has been in an endeavour to minimise the amount of work taken up by the actual driving of the

slide valve. That this work is by no means negligible will be shown in the chapter dealing with valve friction and means of minimising the same.

The first development from the ordinary plain flat type is the valve called the "double ported" one, and shown in fig. 26 in its mid position. The first thing that strikes one is that the valve is very large, and this is due entirely to its being the valve for a low-pressure cylinder, and as is well known these latter are sometimes of great diameter, 5, 6, and more feet, and when we remember that the valve for such a cylinder has a width of about $\frac{5}{8}$ of the cylinder diameter, and a total depth of practically the same, we quite see the valve is bound to be a large one. Its size is not due to the double porting entirely, though it would be possible to make it a little less in depth, but not in its width, if we gave up having what are called the steam pockets *S* and *S* (fig. 26). The object of using these double ported slides is to reduce the travel needed in order to get the large volume of steam into the L.P. cylinder. We must remember that the steam as it comes to this cylinder is weak in pressure, sometimes only about 4 lbs. above the atmosphere, at others perhaps as high as 12 lbs.; and even though the actual *weight* of steam is now less than what the H.P. cylinder exhausted to those following it (owing to engine leaks, cylinder condensation, etc.), yet it is not a question of weight of steam, but entirely one of volume. Taking the steam entering H.P. cylinder as of 175 lbs. gross, 1 lb. weight of it occupies only 2.56 cubic feet, whereas that same weight if only of a pressure of about 25 lbs. gross will need a space of just 16 cubic feet, and given that the speed of its passing through the passages of the L.P. cylinder is half as great again as that of passing into the H.P., yet we have the plain fact that a very much greater area of passage is needed to accommodate this enlarged volume of steam.

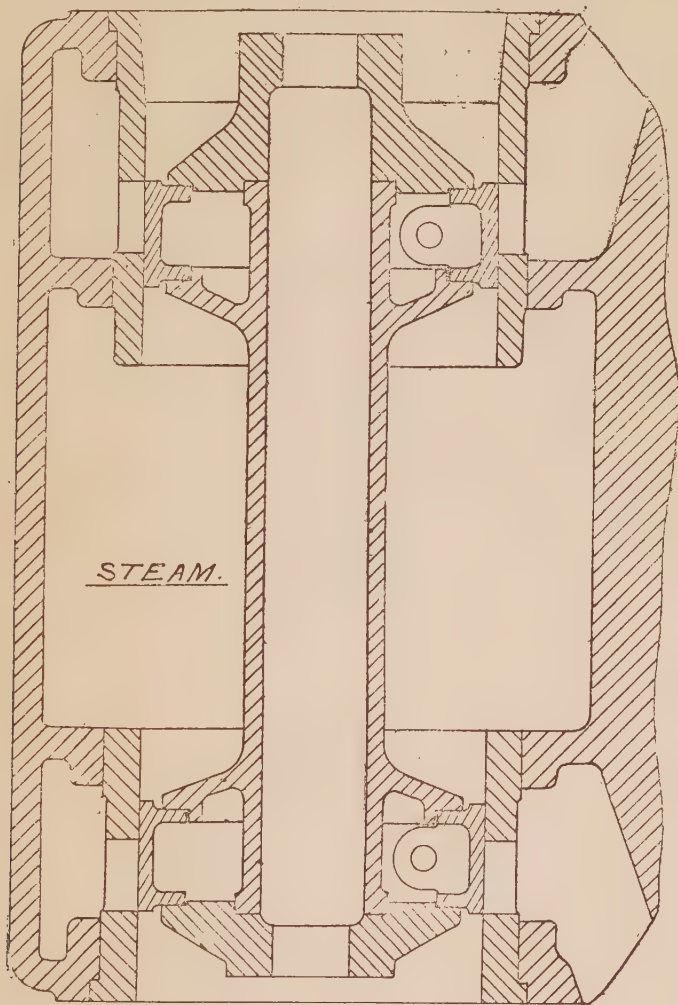
A description of the double ported valve in greater detail is as follows. It is practically an ordinary slide valve only cast with a very deep exhaust cavity, across which are arranged two specially designed *steam* passages or pockets as they are sometimes called. These always have steam in them, it obtaining access through the ends of each pocket being open to the steam in the chest. The function of these pockets is to admit steam to the inner pair of steam ports, seen in the section of the specially designed cylinder face, and it is to be noted that whilst the valve we are describing is usually termed a "double ported" one, that name would really be more accurately describing the cylinder face, which has two distinct steam ports at the top and a similar two at the bottom. As seen, each of these lead or merge in one

another and so only one passage is needed into the cylinder at each end. What is termed the "middle" bar, seen at *A* and *A* in figs. 26 and 27 has to be designed and made of such a depth as will ensure that the valve, in its travelling to and fro over the cylinder face, *never* brings either of the steam pockets open to the exhaust, and should our reader be a candidate for a Board of Trade certificate and called upon to make a sketch of one of these valves and cylinder face, we warn him to draw this middle bar of ample depth, else he will have things arranged that the above mentioned trouble occurs.



These valves are often designed to have minus lap, that is, the edges indicated by *a a* in fig. 26 are arranged such that when the slide is in its mid position as shown, the ports are open a certain extent to the exhaust and so help to free this better, by giving a greater time for the ports to be open to the exhaust. This becomes very necessary if we know that the design is for a L.P. cylinder in which the pressure of

steam is only to be, say, about 5 lbs. above the atmosphere at entrance and to expand down to a matter of perhaps 7 lbs. absolute back pressure, when its volume will be vastly greater.



— *Fig 28* —

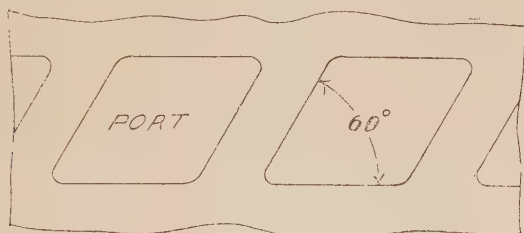
The two figs. 26 and 27 show also very clearly how large the area is at the back of the valve and hence with these double ported slides it is usual to fit some form of "relief ring," which will be described in detail in a later chapter of this work.

Fig. 28 shows a piston slide in a more extended manner than did fig. 24, and the first thing that strikes one on looking at these two is that the piston type evidently costs more to make than the flat form, and this is quite true, but the extra expense is a good deal more than counterbalanced by the easier working of the piston form, less lubrication of its surfaces, less power to drive it, greater lasting properties and, in fact, at the present day, one rarely, if ever, finds a marine engine of anything like large horse-power, being fitted with a flat form high pressure slide valve.

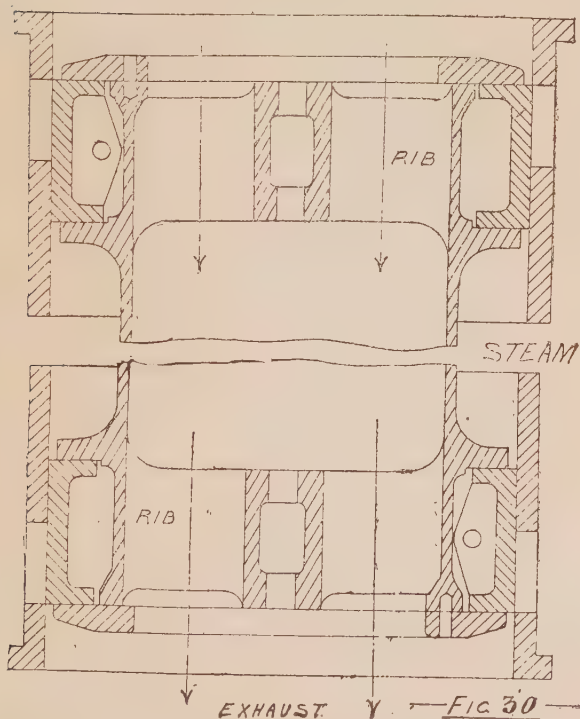
As seen from fig. 28—which, by the way, shows a pattern of piston slide adopted by one of the best engine building firms in Great Britain—it consists essentially of a long central hollow casting, through which the valve spindle (not shown in fig. 28) passes. Towards the top and bottom of this central piece is formed a flange for the reception of the respective rings that form the actual valve portion, *i.e.*, those parts which control the cut off of the steam, and opening and closing to the exhaust. These rings are also of a good quality close-grained cast iron, and very accurately machined to fit the two chambers, or liners, in which they work. Means are provided whereby wear of the rings can be taken up within certain limits, which arrangement, even though these limits are small, still makes for greater steam tightness of this part than the solid ring type does. To keep these “valve rings,” as we may call them, in their place, other castings—which might be termed caps—are fitted to embrace the valve spindle, and when the whole thing, consisting of central sleeve, valve rings, caps, etc., is slipped over that spindle down to a collar there is formed to receive the lower cap, the fitting of a couple of nuts above the upper and larger cap secures the whole firmly in its place. One should note that in turning and fitting the above-mentioned “caps” a clearance is arranged such as will allow the rings to adjust themselves, or, as it is termed, they are made “floating” or partially so. The types of piston slides vary very much, some being even more elaborate than fig. 28, whilst on the other hand there are some makers who believe in what is really nothing more than a solid block. With such as this, one has to depend for steam tightness upon the fact that ordinary steam as it comes from the boiler is in a condition known as dry saturated (and sometimes even wet saturated, such as when priming occurs), and so carries a fairly large percentage of moisture, which, getting on both the face of the valve and the chamber as well, seals the surfaces and so prevents very largely a leakage of steam. Grooves are also turned in each of the “pistons,” and these become filled with this wet, thus

forming a water seal on the same lines as that arranged in an Edwardes air pump. Should the steam, however, be superheated, then none of this moisture being present necessitates special lubrication to effect this

EXPANDED VIEW OF PORTS IN LINER:



— FIG 29 —

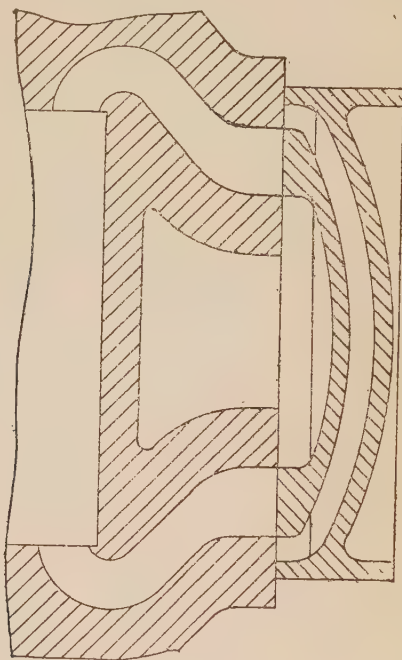


— FIG 30 —

sealing and also to prevent the surfaces "siezing." It will be seen that with fig. 28, the steam being admitted over the inner edges of the valve means that the outer edges constitute the exhaust ones, and as the

central sleeve is not designed to allow the exhaust of one end to pass *through* it to the other, we must arrange *two* exhaust openings in the chest of this cylinder, and by connecting these externally by means of a large diameter pipe we can, by branching off this, lead the exhaust to the next cylinder.

Whilst the figure does not perhaps show very clearly the following, yet careful measurement will reveal the fact that the diameter of the upper valve ring—or, as some term it, the upper piston—is a little more than the lower one, and, as the steam is inside, the effect of this

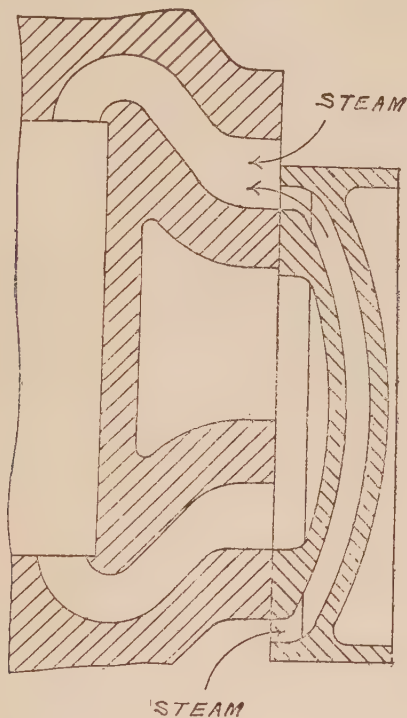


—*FIG 31*—

increase is to make the upward effort more than the downward one, resulting in some of the weight of the valve being taken off the gear used to drive it. This is another excellent feature of inside steam piston slide valves.

From fig. 28 we see that the steam is admitted all round the valve, but as it can only pass into the passage leading to the cylinder at one place, we have to design the chamber portions within which the upper and lower pistons move in such a way as will prevent the valve rings

having a tendency to spring into the steam passage, and also to prevent these rings becoming grooved. To do this the steam passage is cast with a series of diagonal ribs across it, and making an angle of about 60° to the horizontal, as by so placing them there is no fear of a groove wearing in the valve ring. Fig. 29 shows the idea. Fig. 30 shows us a piston slide of a very large diameter, and with such there is plenty of room for the exhaust steam to pass through the central portion from one end to the other, and so there is only the one opening called for



—FIG 32.—

on the side of the chest to lead the exhaust steam to the next cylinder, or, as the case may be, to the condenser.

Summing up the advantages of a piston valve with inside steam they are, less friction and loss of power for driving; less wear and tear on the valve gear; less temperature of the steam coming against the packing in the spindle stuffing box. With outside steam admission these advantages are reduced, being only the less friction and loss of power.

Another form of valve designed to save power in its driving is that called the "Tric," after the name of its inventor. Fig. 31 shows the form, and though it can be arranged as a piston slide, yet the majority of Tric valves are flat. Its peculiarity lies in the steam passage formed in the back of the valve and going from one end to the other. In fig. 32 it is shown with one port full open to steam, and it is to be noted that then a part of the entering steam is admitted over the outer edges as usual, whilst the balance comes up by means of the passageway before mentioned, and so really this valve is a species of double ported affair, and, in fact, is as much entitled to that designation as the ordinary double ported one. It will be noticed that with the "Tric" type the ports in the cylinder face are not altered in number.

With these valves care must be taken to see that the cylinder face is of correct depth to ensure the closing of the steam passage through the valve, at the same time as the steam is cut off at the outside edge of valve in the ordinary way.

With this class of valve (Tric and usual double ported forms) one has to be careful with their setting to see that the leads are kept at the correct amount, because by admitting steam over two edges at once, the *actual* lead is the sum of what there is at each steam edge, or say we want $\frac{1}{4}$ inch lead altogether, then $\frac{1}{8}$ inch would be arranged for at each of the two steam edges.

Remember when drawing a keyway diagram for a double ported valve of the type shown in fig. 26, that the distance "lap + lead" has only to be that which is given *either* at the outside edge of the valve, *or* that given at the steam pocket S. For a Tric valve the distance "lap + lead" is the distance from steam edge of port to outside edge of valve; in fig. 31, plus the lead given *either* at that outside edge, *or* else the amount of lead she gets from the bottom, *via* the steam passage in the back of the valve.

CHAPTER V

OUR reader, by this time having, we trust, a good idea as to slide valves generally and their actions and importance so far as the proper running of the engine is concerned, will doubtless wish to know why it is that economy can be obtained by the usage of steam of a high pressure.

Heat being the source of all work in the steam engine it is necessary to have some agent in order to convey the heat from the coal or oil in the furnace of the boiler, to the actual working parts in the engine. For several reasons we use water to do this carrying, and in order to render it into a useful form we apply the heat to it and convert it into its gaseous or almost truly gaseous form of steam. It is from the aspect of £ s. d.—rather unfortunate—that we should have chosen such an agent as water, because due to its great latent heat there is so much needed to change 1 lb. of it into 1 lb. of steam, that the greater portion of the expense of running a steam set is taken up by this; and further, the latent heat does no really useful work in the engine. Perhaps we had better explain what is meant by latent heat and sensible heat. The latter is what affects the reading on a thermometer; that is, if we were experimenting with, say, 1 lb. of perfectly pure fresh water and by application of heat were trying to evaporate the whole of it away and had placed in this water a thermometer, we should see as the contents of the vessel became warmer that the mercury was, of course, getting higher and higher, until at last when it registered 212° F. the water would be boiling. If now we wish to evaporate the whole of this 1 lb., we shall have to go on putting heat into it, or in other words must keep the vessel over the source of heat for a considerable length of time, and our commonsense tells us that the water is receiving heat all this while and yet inspection of the mercury in the thermometer shows it to be *no* higher than when the water was just at the boiling point, so we see that the heat passed in to produce actual evaporation, whilst being of a greater number of thermal units, is yet of

no higher intensity, and we, therefore, call this latter "latent" heat. Put briefly, this is the heat that changes the state or form of the agent without altering its temperature.

It is sensible heat that does work in the engine, by which we mean that work is done by a fall in temperature of the steam. If the latent heat is parted with as well, then the steam returns to its original condition, viz., water.

The thermal units (T.U.s) required to evaporate 1 lb. of fresh water at a temperature of 120° , which may be taken as an average for feed not separately heated, are given by $1115 + 3 \times T^{\circ} - t^{\circ}$, where T is the temp. of the steam corresponding to the pressure under which it is generated, and t is the temp. of the water at first, *i.e.*, the feed.

Suppose we are going to use a low pressure of, say, 40 lbs. gross, for which the temp. is 267° in round figures, then $1115 + 3 \times 267^{\circ} - 120^{\circ}$ or 1075 T.U.s per lb. will be required. Next, suppose we wish to use steam of four times the pressure, or 160 lbs. gross, for which the temp. is 363° , then $1115 + 3 \times 363^{\circ} - 120^{\circ} = 1104$ T.U.s per lb., showing that only $1104 - 1075$ or 29 T.U.s more are needed to produce four times the pressure. It is, of course, clear that this must cost more expenditure of fuel to produce these extra 29 thermal units, but remember we shall not require anything like so much steam of this higher pressure to give us the same horse-power as if we used steam at the 40 lbs. gross. This we now proceed to show.

As data, take two engines both to indicate the same horse-power, at the same revs. per min. and of the same stroke. One uses steam of 40 lbs. gross pressure, whilst the other uses 160 lbs. gross and absolute back pressure of 3 lbs., and for simplicity let us take the steam as being used in only one cylinder. In order to get the desired amount of work, we may have in the first case to carry the steam for possibly $\frac{3}{4}$ of the stroke, thus exhausting it theoretically at 30 lbs. ($40 \times \frac{3}{4} = 30$). Taking a stroke of 4 feet and a diameter of 2 feet, we have for the volume of that cylinder up to cut off, $2^2 \times 7854 \times 4 \times \frac{3}{4} = 9.42$ cubic feet to fill with low pressure steam. The specific volume, or the volume in cubic feet of steam at the given pressure, from 1 lb. of water, is for 40 lbs. gross, 10.4 cubic feet, and so we require $9.42 \div 10.4 = .905$ of a lb. of steam or $.905 \times 1075 = 972.8$ T.U.s in this case. Calculating the mean pressure of this steam from the forl. $Pm = P_1 \frac{1 + h \cdot \log. r}{r}$; where Pm is the absolute mean pressure, P_1 = the absolute initial pressure and r = the number of expansions of the steam, and which are found from $1 \div$ cut off fraction. We, therefore, have for this case $1 \div \frac{3}{4} = \frac{4}{3}$

or 1'33 as r , and so $Pm = 40 \frac{1 + h \cdot \log. 1'33}{1'33} = 40 \times '965 = 38'6$ lbs., and from which the abs. back pressure of 3 lbs. must be subtracted, making the mean *effective* pressure 35'6 lbs.

If now the horse-power of the second engine using 160 lbs. steam has to be the same as before, we must have the *same* mean eff. pressure, seeing that the rest of the data remains unchanged, hence $38'6 =$

$160 \frac{1 + h \cdot \log. r}{r}$ and we must find r , which comes out at practically

15, that is, there will have to be 15 expansions of the steam now, or cut off occurs at $\frac{1}{15}$ stroke. This is, of course, much sharper than could be got with an ordinary slide valve, but an expansion one could give it, although for the moment we do not bother about considering this point, our sole aim being to show the saving by high-pressure steam. $\frac{1}{15}$ of 4 feet is '266 feet, so only '266' $\times 2^2 \times '7854$ or '8356 cubic feet require filling. Then, as the specific volume of steam at 160 lbs. gross is 2'8 cubic feet per lb., we have '8356 $\div 2'8$ or '298 lb., and so '298 $\times 1104 = 329$ T.U.s (in round figures) are needed under our new conditions, showing a saving of $972'8 - 329 = 643'8$ T.U.s, proving conclusively that high-pressure steam shows a great gain. In every day practice it would not be quite so large as has been proved here, but even when all allowances have been made for condensation, radiation, etc., there still remains a very marked gain.

At this point we wish to draw attention to what is very often referred to erroneously as "the gain being due to expanding the steam." That the benefits attendant on using high-pressure steam are not in any way caused by the greater number of expansions should be quite clear from what we have written, but in case this point is not even yet as plain as might be we explain further.

As the engines do not require such a weight of steam when this is at the high pressure, it is clear the boiler need not produce so much, and hence we can do with smaller boilers; and this is exactly what took place in the past days when two-stage compounds displaced the single-cylindere engine, and again when the triple expansion was introduced as an advancement on the two-stage engine. So as less coals are burnt we think it is very evident the gain is due to the high pressure and not to expansion. What this latter really does is this. If after saving in the boiler we were to use steam recklessly and wastefully in the engine we should simply be throwing away all gains and so adopting a ridiculous policy, and it is in order to use the high-pressure steam to the *best* advantage that it must be cut off at some earlier part of the

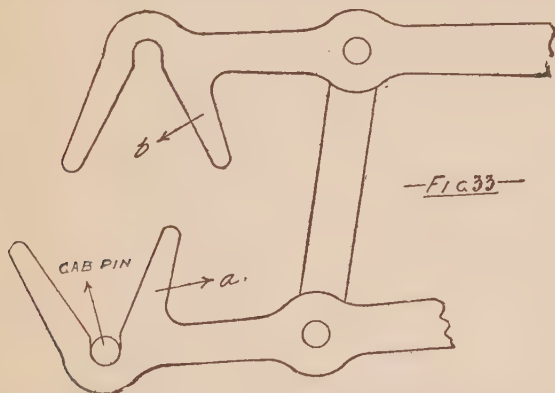
stroke and allowed to expand, fall in pressure (and temperature), and so go to the exhaust at an economical pressure, which in condensing engines will be somewhere about 10 lbs. gross.

In order to prevent such very sharp cut offs as that which was required in the case just quoted, viz., $\frac{1}{15}$ stroke, and which, *if given*, would mean a very great variation in the load coming on all the moving parts, we make the diameter of the H.P. cylinder less than before and so with all such as stroke and revolutions remaining constant we can allow a greater mean effective pressure to act on this smaller piston, and thus keep the horse-power the same.

CHAPTER VI

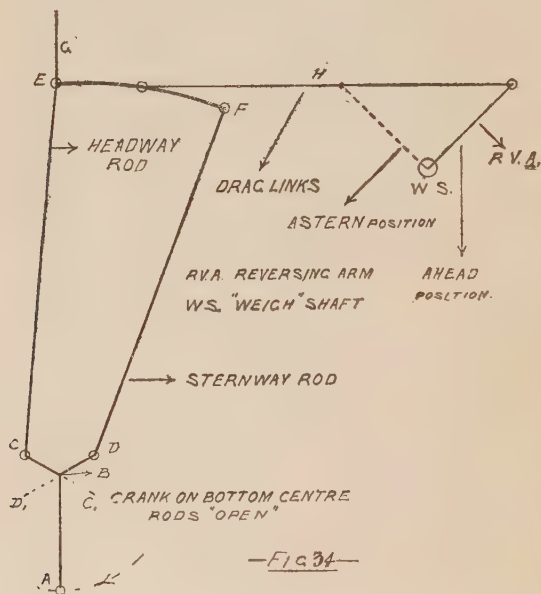
IN this we describe the link motion necessary with a reversing engine, and show the benefits to be obtained by such. We introduce "linking up," and point out the effects produced when this is resorted to. Readers are advised to very carefully follow this part as it forms the most important portion of what we shall describe a little later in this book, viz., the balancing or adjustment of the horse-power between the different cylinders of an engine.

In days long gone a valve gear was used on reversing engines, which consisted of two eccentrics, straps, and eccentric rods, just as we have to-day, but with the important difference that the ends of the eccentric



rods were not secured to the ends of a curved reversing link as we know them, but had large forks formed for the purpose of dropping over a gab pin placed on an arm which drove the valve spindle. Fig. 33 shows the idea, and to reverse it was only necessary to move over a hand lever, which brought fork *a* away from the pin, and at the same time fork *b* came down, and directly either face of that fork touched the gab pin it moved it along in such a direction that the steam was admitted to the opposite side of the piston, and so the engine reversed.

As soon as steam pressures grew it was recognised that to obtain economy it would be advisable to have some means of altering the cut off to suit varying conditions of road, as it must be remembered that the arrangement we are describing was fitted to locomotives. It was at once seen that the gear in fig. 33 was quite unsuitable, as the instant the fork was moved off the gab pin this latter ceased to be controlled for at least a part of the engine's revolution, and so stoppage would be brought about. Many turned their attention to this matter, and about the year 1842 a fitter, named Howe, designed a gear in which the two eccentric rods were connected together by a slotted link, and by attach-



ing the valve spindle to a block which was placed in the slot of this link he was enabled to move the eccentric rod end away from directly in line with the spindle, and thus have what is termed "linking up." By so doing the travel of the valve is always shortened and the cut off is made earlier, but these points we shall explain very shortly in greater detail. Whilst Howe is without any doubt entitled to the whole claim of the inventor of a link motion such as is met with in thousands of engines at the present day, yet the works at which he was employed have given their name to it, and one knows the gear to-day as the "Stephenson" link motion. We are not engaged in an historical controversy, but the

writer knows the facts above to be correct, and so we will proceed to a more detailed description of this almost universal gear.

As all engineers are acquainted with eccentrics, straps, and rods, we only show in fig. 34 the skeleton, as it were, of the motion, and for that reason it becomes all the easier to follow. AB is the crank, shown on its bottom centre, which is the position favoured by most writers on valve motions. BC and BD show the two lines of throw of the eccentrics. CE and CF represent the eccentric rods, it being understood that these are attached to their respective straps, which embrace the headway and sternway sheaves. EF shows the curved reversing link, which to-day consists generally of two solid forged curved bars connected together by distance pieces, bolts, and nuts, the whole forming what is termed a "double bar" link, and allowing space for the tumbling block and spindle to come down between the bars. G shows the slide valve spindle, and which is attached to the "tumbling" block over which the link EF can be pushed and pulled by the "drag" links H , these being links going sometimes from one of the eccentric rod forked ends or, as set up by other makers, from the centre point of the reversing link, and leading away to the reversing arm on the weigh bar shaft. In fig. 34 it is to be noted that the eccentric rods are quite clear of each other, *i.e.*, they are said to be "open" rods, but remember that when the crank turns to its top centre, that D comes round to D_1 and C to C_1 and so the rods then "cross" each other and this opening and crossing is a point we want to bear carefully in mind. The arrangement shown in fig. 34 is one where the eccentric rod end can be got directly in line with the slide spindle, but very often the attachments to the reversing link are made at its extreme ends and then we can no longer get rod and spindle under one another.

The length of the eccentric rods for such a valve motion should be as great as practically possible, but about eight to ten times the valve travel is a fair average, this length being reckoned to the centre of the eccentric sheave and measured up to the centre of the forked rod end. The curvature or radius of the reversing link EF is equal to the distance between the centre line of the link EF and the centre of eccentric. The length or distance on this link between the two points where the eccentric rods are attached varies a good deal with different makers, but as an average we may take from two and a half to three and a half times the travel of valve.

If much variation in speed and power by means of "linking up" is required, then a long link must be used, but if we are content to only have very moderate variation, a short link will do all right

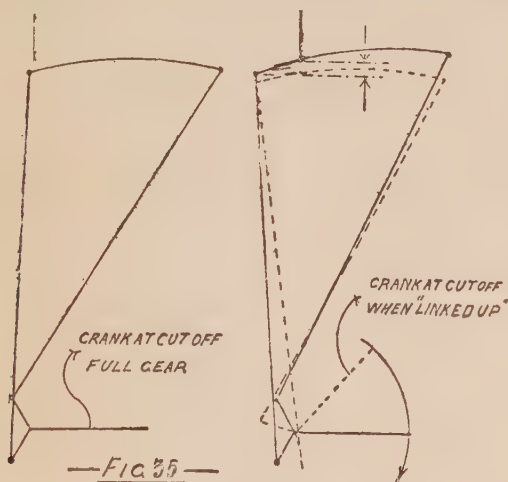
Reversing links are at the present time almost universally made curved, but from the point of view of merely obtaining a means of reversal of the engine, without losing control over the valve, there is no reason why a straight link should not be employed. In fact, even yet some locomotives use a perfectly straight one. Certain considerations outside the scope of this work, however, dictate that it is better to use the curved pattern.

The actions of a link motion such as is indicated in fig. 34 are as follows. Taking the crank on its top centre, and the rods then to be "crossed," the valve has top port open for lead, and as the line of throw of eccentric is ahead of the crank the valve moves down. At the same time the other end of the link is under the influence of the sternway rod, etc., and is going up, and remember the link as a whole is compelled to move in an arc of a circle, whose radius is the length of drag link *H*. On the engine moving, the headway end of link comes down, whilst sternway end rises, and at the same time the link and rods are pushed over a little by the swinging of the drag link, resulting in the end of the headway rod being taken away from directly underneath the spindle. This is augmenting the slotting action, which is the name given to the movement of the link in a direction more or less at right angles to the vertical centre line of the engine, and which is manifest to anyone watching a link motion working, and, in fact, is really the greater source of this feature, which we would gladly do away with if possible. As the crank continues to turn, a point is reached at which both eccentrics are rising and so lifting the link bodily. Then shortly afterwards the headway rod alone goes on lifting the link and spindle, and the sternway rod proceeds to pull down its end of the reversing link, and so the drag link point of attachment still acting as a fulcrum, slotting motion again ensues to a pretty large extent. When the crank is upon the bottom centre the headway rod has risen sufficient to open bottom port for lead, and the engine starting its return stroke the headway rod goes on rising for a time opening port to steam, whilst the sternway rod pulls down its end of link. In a short time we notice that both eccentric sheaves are pulling the link downwards, and that the slotting action has in consequence practically disappeared for a brief time. But shortly we observe the sternway rod going up again, whilst the headway one comes down so that the valve may be brought to the correct position for re-admitting steam to the top port.

It will be noticed that the slotting motion effects to some extent the same thing as though we had definitely moved the headway rod from directly below the valve spindle. We cannot stop this action entirely,

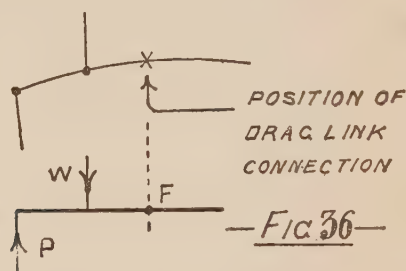
but it may be minimised by having the valve travel as little as possible and the drag links as long as practicable. To allow this slotting motion to be carried out and not to bring the end of link hard up against the side (or end) of the tumbling block, the actual length of the curved portion of the reversing link is made a fair amount longer than might at first appear to be necessary, and so in fig. 34, whilst apparently the ends of the eccentric rods go straight to the ends of the link, yet the link itself is longer. In skeleton diagrams such as these it is not the custom to show this extension, as it in no way affects our explanation of the different functions.

Referring now to what linking up primarily establishes, viz., sharpening the cut off of the steam, we must turn to fig. 35, which



gives in skeleton form the position of the different parts both in full gear and when linked up; crank lying horizontal when cut off is assumed to occur. The full gear one shows how the sternway rod has its end of the link elevated, and so when we run the link over a little we of course bring a part of this higher situated portion under the spindle and in consequence this latter is bound to rise. If, now, we imagine the spindle to be still in its *old* position, then under the linked up condition the engine will not have turned so far when, under the action of the two eccentrics, the reversing link is brought to its now slightly lower position. This means the piston has not travelled so far on its stroke, and, therefore, cut off is earlier. This applies both for "crossed" and "open" rods.

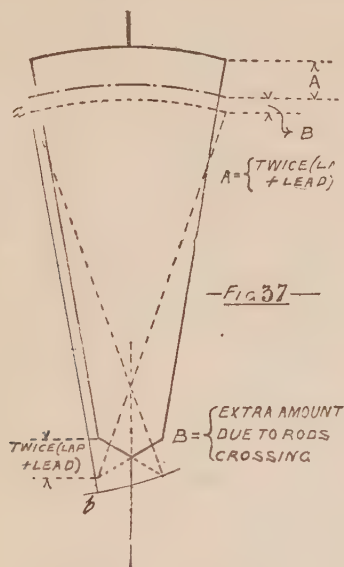
The next feature of linking up to be considered is the reduction in the travel of the valve when the gear is in any position other than "full gear." When experimenting with a model slide valve motion we find that as the link is moved more and more into a position such that the tumbling block comes midway between the two eccentric rod ends, that the travel of the slide valve gets less and less, and, without going into any rather abstruse reasoning to account for this, we may liken the reversing link to a lever with its fulcrum and points of application of force and weight. Thus the place where the drag link is attached is the fulcrum, where the eccentric rod end which is in gear is attached is the point of application of the force which is here the effort of the sheave driving the valve, and the place where the tumbling block happens to be is the point of application of the weight, or, better still, the point of the resistance to be overcome. The lever is, therefore, one of the "second" order, see fig. 36, and the study of levers shows



us that under such conditions, if P only rises for a certain distance that remains constant, W will rise for a *less* amount, and as, actually, our link both rises and falls as the point P travels the amount or distance due to twice the line of throw of eccentric, then it is very clear that W or the spindle will under the "linked up" conditions move through a *less* distance, therefore the valve travel is less. We neglect any effects from the action of the sternway rod and their angularity. Actually, when the reversing link is moved into the mid position, the travel of the valve is so much reduced that neither steam port is opened sufficient for the engine to revolve, and so if we desire to prevent its movement at any time simply middle the link and it is held.

If the valve is of the elementary type then with the link middled as above there will be *no* travel of the spindle, owing to the fact that as one end of the link goes up the other comes down, because now the two lines of throw never come into the same half of the circle of travel, thus they never *both* act to lift or to lower the link.

When lap and lead are given, however, a different state of affairs exists, because now the two lines of throw both come in the same half of the travel circle, which means that when the crank has turned from the bottom to the top the centres of the eccentrics have moved vertically a distance equal to twice the (lap + lead), and so the valve is moved for this same amount. But that is not all, and we must allow for the fact of the rods changing from open to crossed as the crank goes from the bottom to the top centre. Turning to fig. 37, we see by



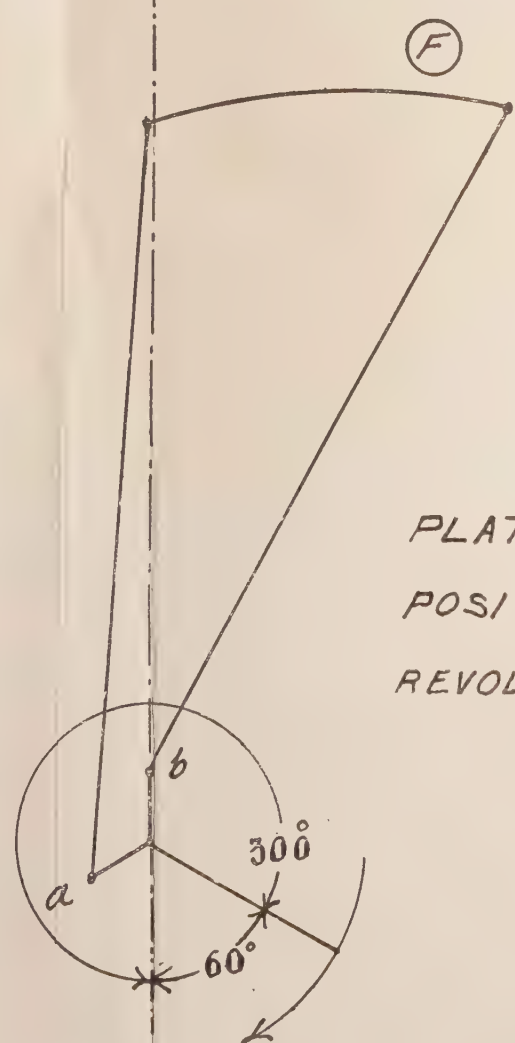
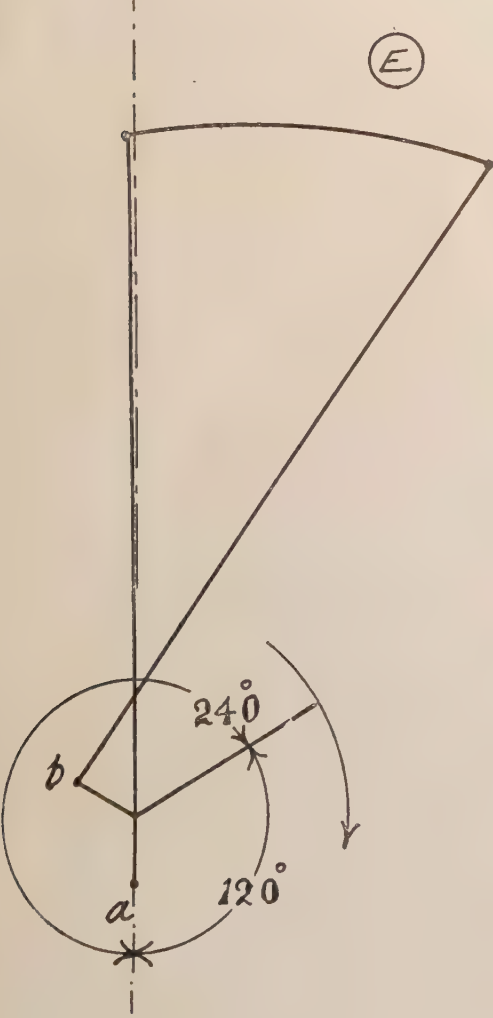
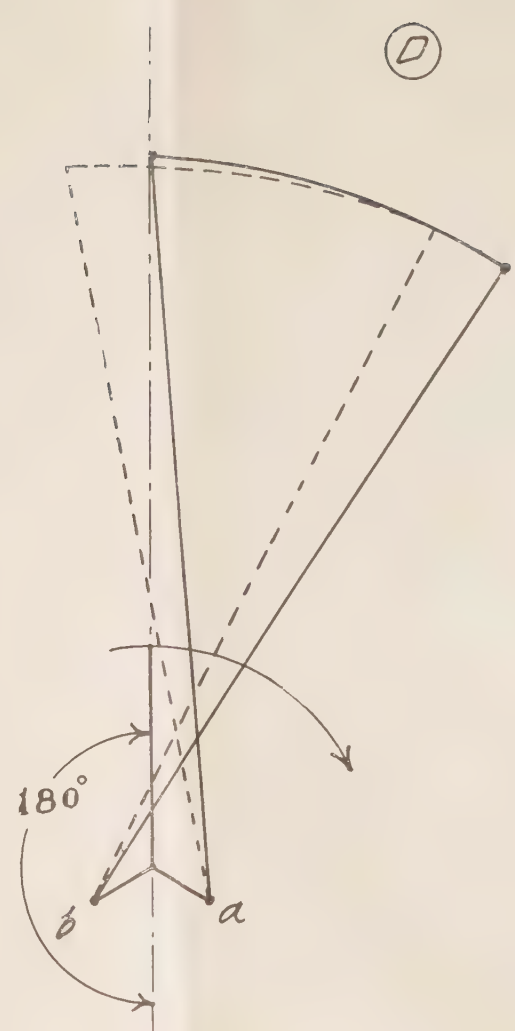
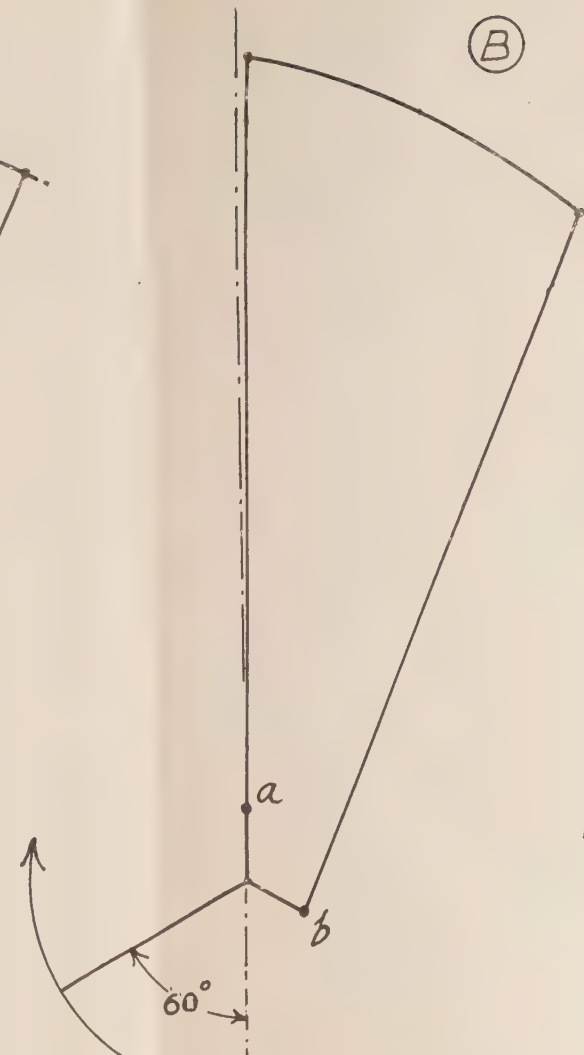
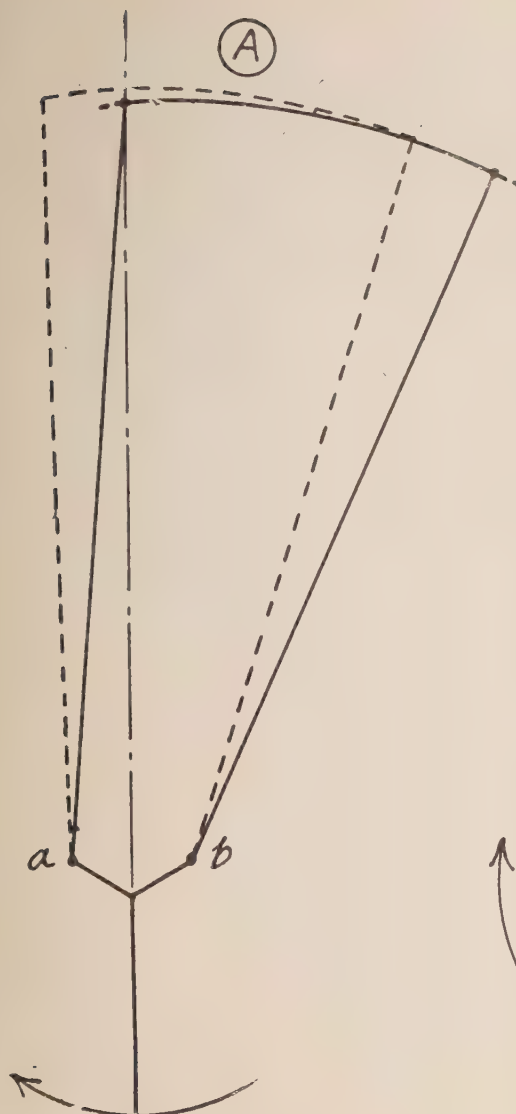
the full lines the position of crank, eccentric rods, link, etc., when the former is on the bottom centre, and by the dotted lines, the new positions when the crank has turned through 180°, *i.e.*, on to its top centre, when it has the eccentric rods "crossed." Observation shows us that the link is now a *greater* distance below its former position than what merely twice (lap + lead) gives, and we wish to know how this is brought about. The reply is that it is due to the increased angularity of the eccentric rods when lying in their "crossed" position. The thin full line *ab*, drawn parallel to the left hand eccentric rod, shows the position that rod *would* have had could it be disconnected and swung round upon the end of the link, thus showing that it would then be a good deal lower, and exactly as much lower that it is, is the amount the link is drawn down below the position for *A*, fig. 37. If

- the rods are put up "open" when the crank is on top centre they become "crossed" when the crank has turned to the bottom centre, and then the movement of the link is twice (lap+lead) minus this amount we have just explained.

The next point in connection with the subject of "linking up" is the alteration in the leads, and in what follows remember we are speaking for outside steam admission. If, however, the steam be inside, all we need do is to think that if at first the change due to linking up was such as to make the leads greater, then with the inside steam these will be the opposite, viz., made *less*. Taking the rods "open," with the crank on bottom centre, or "crossed" when on top, the leads are made more top and bottom, by linking up, for the following reasons. As the link comes over in the process of linking up it brings the headway eccentric rod over to a greater angle, resulting in the link being lowered (see dotted link in diagram *D*, Plate I.), and so the top lead becomes more. When the crank has turned to its bottom centre, the rods being then open, the effect is to lift up the spindle, and so the bottom lead becomes more (see dotted link in diagram *A*, Plate I., and the following description).

Description of Plate I.—Referring only to the full line diagrams for the present, *A* shows the crank on bottom centre and the headway line of throw denoted at *a*, and it is to be observed that the rods are now "open." *B* shows us the position when crank has turned through 60°, and we see at once a great change in the position of the link, caused by, firstly, *a* pushing its end up, and at the same time *b* brings its end down. Slotting motion having ensued, calls for the actual length of link to be such as will enable it to follow the path described by the end of the drag links (not shown). *C* shows how the rods are beginning to cross each other as crank turns still more to the top, and *D* shows the position when crank has finally reached this point, and observe the rods are now "crossed" instead of open. The link also is coming to a more horizontal position as *a* is drawing its end down, while *b* elevates the other end. *E* shows the crank in a position at 240° to the bottom centre, in the same direction as before (or 120° *short* of that centre), and we see the link has come back, and by reason of the slotting motion is again taking the eccentric rod end from under the valve spindle. Diagram *F* shows the rods once more "open" and the parts all returning to the positions they had at first with crank on bottom centre.

The alteration of leads is unfortunate, as very often we might resort to linking up an engine and so controlling its power in a better manner, yet because we may know it will not stand any alteration in its leads we



THE DIAGRAMS SHEWN HERE
ARE IN CERTAIN DETAILS RATHER
EXAGGERATED FOR THE SAKE OF
CLEARNESS.

PLATE I. SHEWING VARIOUS
POSITIONS OF LINK DURING ONE
REVOLUTION OF THE CRANK.

are prevented from obtaining the gain. The only gears which permit of being linked up without affecting their leads are the single-sheave gears, for a description of which see their special chapter in this work.

Other points altered by linking up are those of exhaust opening and closing, and it is enough to state what changes are brought about without going deeply into the why and wherefore. With "crossed" rods, crank on the top centre, the exhaust opens earlier and closes earlier, whilst with "open" rods, crank on top centre, the exhaust opens later and the closing is not seriously altered.

The following table gives the various effects in a concise manner, and must be memorised by Board of Trade candidates for first class.

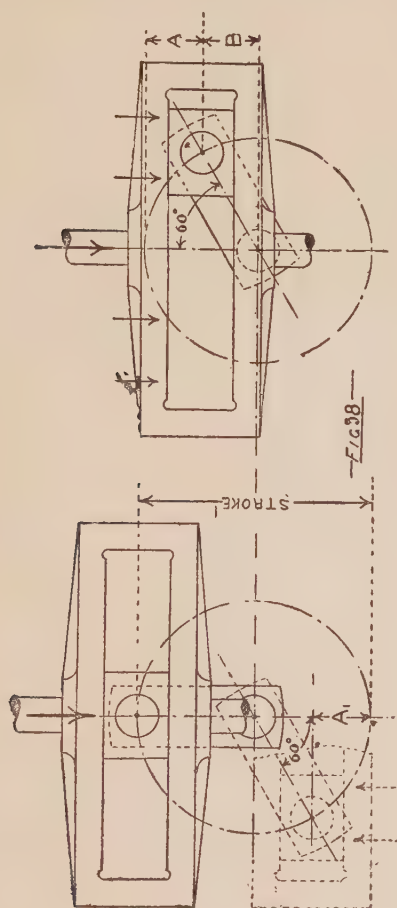
Linking up with—	
Crossed rods gives	{ More lead top and bottom ; sharper cut off ; earlier opening and closing to exhaust ; less travel
Open rods gives -	{ Less lead top and bottom ; sharper cut off ; later opening to exhaust with closing of exhaust not seriously altered ; less travel
In above the crank is considered as on top centre.	

CHAPTER VII

THIS deals with the difference existing between the cut off from the top and from the bottom of the cylinder, and explains why a difference exists. To many it would seem as if, when a structure like an engine, say a marine one, is erected and all the parts work properly and without yielding in their shape, that whatever cut off is arranged for from the top downwards that it should be the same from the bottom upwards. But such is not the case in any engine having a connecting rod of a definite length, and whose slide valve has not been specially designed with respect to different amounts of steam lap top and bottom.

We have already seen in fig. 37 the effect of a rod being placed at an angle instead of perpendicular to what it has to control, and whilst in ordinary cases of eccentric rods their length is so great in comparison to their total movement that differences in position of link due to angularity of these rods can generally be neglected, especially when in full gear, yet it is otherwise with the connecting rod and the point of cut off, as here even the longest rod used is as a rule not more than about six times the crank length, and for marine engines is generally only about four cranks, and in small electric light engines the connecting rod is at times down to as little as two and a half times the crank. Our remarks here are mainly directed to an ordinary marine job, but they must be understood to apply to all reciprocating steam engines with a finite length of connecting rod. The only case where such an engine could have equal cut offs without altering her slide valve or introducing a specially-designed rocking lever between the eccentric rod and valve would be one having what is called a connecting rod of *infinite* length, *i.e.*, a "donkey crosshead" fitted in place of the usual rod. Fig. 38 shows one of these, and it will be seen that no connecting rod as we know them exists, its place being taken by a parallel-sided slot link, in which a block carrying the crank pin works to and fro as crank revolves. The line of pressure from the piston acts vertically through the piston rod on to this slot link (or donkey

crosshead), and so the whole of this is being either pushed down or pulled up *in a vertical direction* the whole time the engine is running, and hence the crank pin is always acted on by a force parallel to the axis of the engine itself. When this is the case a line drawn from the centre of crank pin (wherever it may be) on to the centre line through

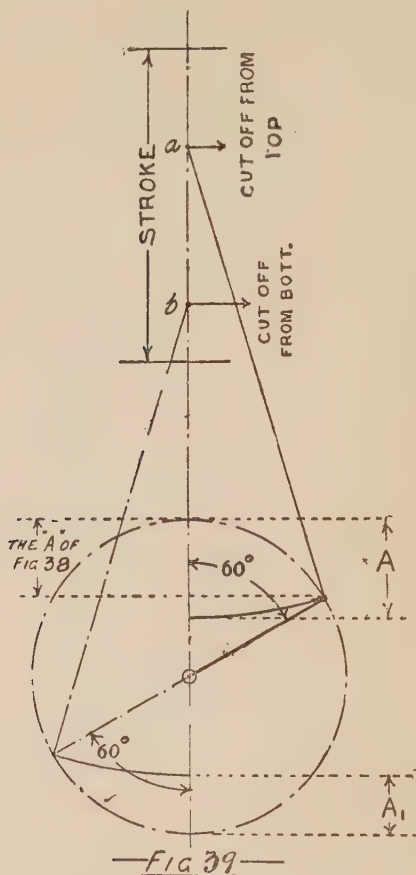


piston rod will show the exact piston movement corresponding to the circular or angular movement of the crank, and if this moves through 60 degrees then we shall see the piston has moved through a distance *A* inches up to point of cut off, and should be carefully compared with fig. 39, and the difference noted. To those understanding trigonometry we

have a ready means of calculating how much A or B is, provided we know the angle of crank at cut off and the stroke.

θ = angle at cut off, r = crank" = $\frac{\text{stroke}''}{2}$, A = versine θ , B = cosine θ .

To express A as some definite amount it is $A = r'' \times \text{versine } \theta$, and, as the versine of an angle is $1 - \cos. \theta$, we have $A = r (1 - \cos. \theta)$. Similarly $B = r \times \cos. \theta$.



Notice also from the dotted line portion of the left hand part of fig. 38, when the crank is coming up, i.e., piston is upon its return or up stroke, that when the angular distance between centre line of engine and crank is the same as it was for the down stroke (here it is 60

degrees) then exactly the same amount of that second stroke has been done as was of the first, or A_1 equals A .

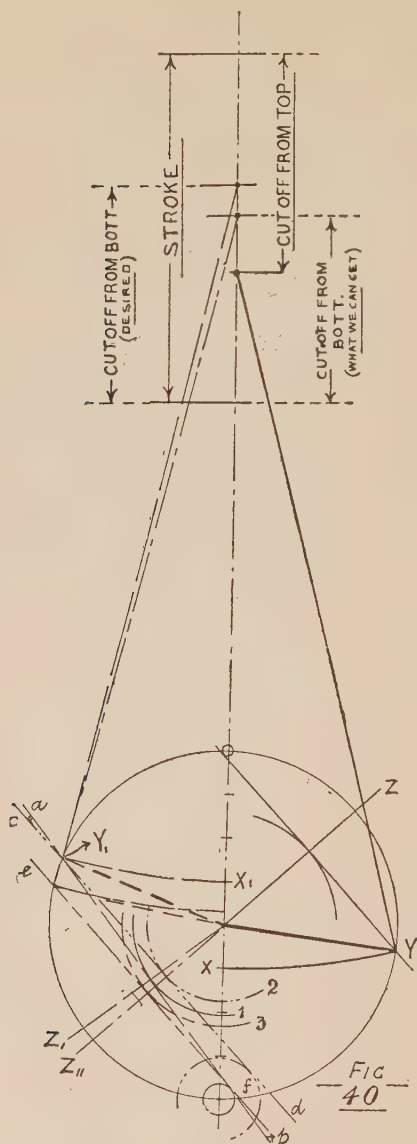
This point also should be compared with fig. 39, and the differences noted. In fig. 39 we have the more practical case of an engine with a connecting rod of a definite length, and in order to bring out the effects in greater prominence we take a rather short rod of only 3 cranks in length. The construction of this figure is as follows:—

Draw the crank pin path as a circle, and for comparison make its radius equal the crank length of fig. 38, that is if one were drawing the two diagrams out on a larger scale. Actually, here, fig. 39 is reproduced on a *larger* scale than is fig. 38. Then lay off crank lines at 60 degrees to the axis of engine as shown. Next, with a length equal to three times the crank, mark points a and b , which give respectively the cut offs from top and bottom of the stroke. At once a great difference between them is noticed, and also that neither A nor A_1 are now equal to what they were in fig. 38. Further, we see the cut off from top is greater than that from the bottom, and this is opposite to what we would have liked, and from a theoretical point of view to make them equal, or to perhaps make the bottom cut off the greater, we must give less steam lap on the bottom edge of slide valve, and so counteract the effect of the connecting rod.

Without going into any mathematical reasoning as to why a difference exists between top and bottom cut offs, we can yet show plainly that with a rod of a definite length such must always exist. Notice in fig. 39 that when the connecting rod swings into the axial line of engine that its bottom end is *nearer* the bottom of stroke than what it is when coupled on to its crank pin, and as the point a corresponds to the position the crosshead and piston are in when the rod was so hanging, we see that a is drawn down below its true position. But this is not enough, for consider the up stroke and bottom cut off. Here as the rod swings into the axial line its bottom end is again nearer to the bottom of the crank path, but observe that now this bottom point is really the starting point of the up stroke, and so the swing of the rod brings point b nearer to the beginning of this stroke, thus making A_1 less than A .

We should now consider what is to be done in order that the two cut offs may be, if possible, equalised, and, if this cannot be done, how near to that desired result we can come.

Fig. 40 shows us the same kind of diagram as did fig. 39, only it has in addition sufficient lines, arcs, etc., to make it what is called a lap finding diagram, and as such is important to all candidates for Board



of Trade certificates. The large circle is made of a diameter to represent the travel of valve to any particular scale, and also the stroke of piston to a different and smaller scale. The vertical centre line is divided into any number of equal parts, usually eighths or tenths.

The small circle at the top has its *radius* made equal to the top lead. Fixing upon a certain cut off of, say, $\frac{5}{8}$ stroke, let us first of all work to get both the top and bottom cut offs the same (if we can), and the construction then of fig. 40 goes on as follows. Taking a connecting rod of a length equal to four cranks, place the pencil leg of the compasses on the $\frac{5}{8}$ mark X , and with them opened out to the length of connecting rod, on the same scale as the stroke has been taken, see where the needle-pointed leg rests on the vertical centre line. This point denotes our cut off from top. Keeping the needle point thereon, swing in the arc XY , and draw the thick full line from Y to the centre of travel circle, and which line shows the position of crank at cut off from top. Now draw a diagonal line from Y tangent to the *left hand* side of the upper lead circle, and then from *centre* of travel circle draw a perpendicular to touch this diagonal, and the length of which perpendicular is the lap needed to cut off steam at $\frac{5}{8}$ of the down stroke. Now, according to the notation adopted when dealing with Zeuner valve diagrams, this perpendicular line is considered as though it were the position of the eccentric's line of throw, and which latter, remember, remains the same both for down and up strokes of the engine.

Now let us turn to the bottom, or next consider the up stroke cut off and the bottom lead. We set out to see if the two cut offs *could* be made equal, *i.e.*, we wish a $\frac{5}{8}$ stroke cut off going up. With our compasses opened and set exactly as they were before, place the pencil point upon the $\frac{5}{8}$ mark counting from the bottom, or X_1 , and see where the needle point rests now upon the vertical centre line, and with that point fixed swing in the arc X_1Y_1 . As the lap line of the down stroke part (Z) is taken as position of line of throw, whatever angle it makes to the vertical will be the same angle as its continuation on to Z_{11} makes with the vertical on the bottom half, and so a diagonal cd from point Y_1 , to be drawn such that ZZ_{11} may be *perpendicular* to it, will mean a tremendous bottom lead, as shown by the large partially completed lead circle. Actually it worked out for fig. 40 at just 1 inch, which in comparison with the $\frac{1}{8}$ inch lead at the top is absurd. Further, the lap needed for this bottom cut off is also very much less than that for the top, and the difference between both the two leads and the two laps is far and away more than the engine would work with, and so we must dismiss the idea of getting the two cut offs exactly equal.

In fig. 40 we have drawn a diagonal line ab tangent to a lead circle having only $\frac{3}{8}$ inch lead, which is a quite permissible one, but now note the following carefully. As this new diagonal does not run parallel to either cd or yet the first drawn diagonal one (for the down stroke), it is

evident that to get a lap line perpendicular to ab it will need turning a little until it gets into position Z_1 , and now $Z Z_1$ are not in one straight line, which shows that we should have to move the line of throw of sheave, in other words, this latter would require turning on the shaft, and at once this upsets the top setting. So we see we cannot have *equal* cut offs top and bottom and a reasonable lead at the same time.

What we finally do is this. We keep the moderate lead, say the $\frac{3}{8}$ inch, and draw a diagonal line ef , which shall be parallel to the first drawn one, and which will enable the one setting of the eccentric to suffice, and then where this diagonal cuts the travel circle place the pencil point of the compass and see where the needle leg rests on the vertical centre line. It will be lower than before, but not so low as if we had had the same lead at bottom as at top. Note also that the bottom lap is not as great as the top amount.

We have already said that fig. 40 can be used as, and really is, the lap finding diagram, though for the ordinary examination conditions we do not consider the connecting rod, and so, having drawn the large circle to a suitable size that it may represent the travel of the valve, we can proceed thus. Suppose an examiner desired to know from a candidate for, say, his second-class certificate how much lap would be needed to alter a cut off from $\frac{3}{4}$ to $\frac{1}{2}$ stroke, then draw the vertical centre line and divide into four equal parts. Draw the small circle at the top with its *radius* equal to whatever lead we choose to assume. Next draw horizontal lines through the $\frac{3}{4}$ and the $\frac{1}{2}$ stroke divisions until each cuts the large or valve travel circle. From these two points next draw diagonals so as to be tangents to the lead circle periphery, and, lastly, from the *centre* of the whole figure draw two shortish lines, each perpendicular to one of the diagonals, and these last drawn lines are, as seen from fig. 40, the measurement of the respective laps for the two cut offs.

As the length of the lap line for $\frac{1}{2}$ stroke cut off is more than that for $\frac{3}{4}$, it follows that the *difference* between the two lengths is the additional lap required to be pinned on to the top steam edge of slide valve (if we have used the lead at the top port), and then in order to regain our original lead, seeing that the above extra lap has probably taken it all away or at least has seriously diminished it, we must advance the eccentric a sufficient amount further round the shaft.

As the bottom lead is never the same as the top one, we should by rights construct another and similar lap finding diagram for the bottom cut off, etc., but it usually suffices for the examination if it be done for the one port only. Readers' attention, if they are going in for their

second-class certificate, is strongly needed on this matter of the lap finding diagram, and they should practise drawing the same for several different cases of alteration, and amongst these let them take one or two where the cut off has to be made *later* instead of earlier, and in which case we construct the diagram exactly the same, only remember that now the lap difference found has to be taken *off* the valve at its steam edge, and the sheave is put *back* to regain the old lead.

CHAPTER VIII

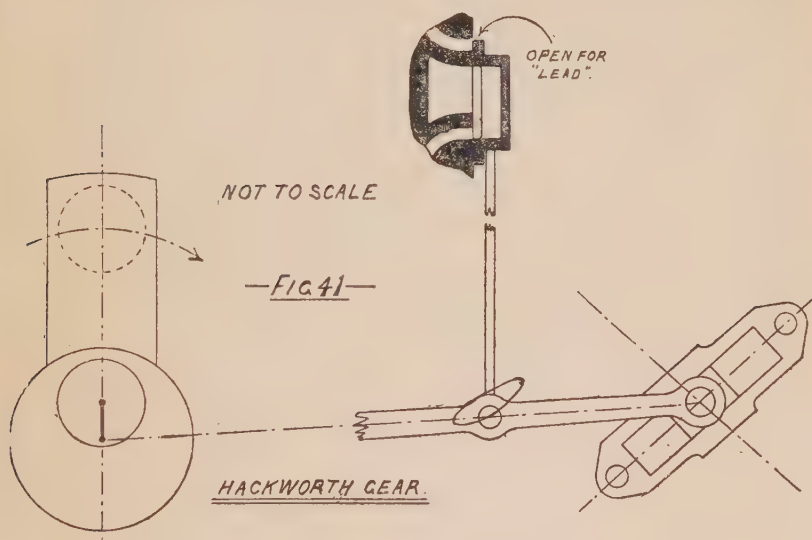
AS questions crop up at examinations regarding patent valve gears, and ones using a single eccentric, or none at all, or one loose upon the shaft, also the means adopted to obtain very sharp cut offs, we give the following descriptions of such as Hackworth's parallel slide single sheave gear, Marshall's (or Bremmes') swinging link single sheave motion, Joy's gear worked off the connecting rod of the engine itself, Meyer's separate expansion valve, and the old, but very useful, loose eccentric.

We stated in Chapter VI. that one of the evils of the Stephenson link motion manifested itself when the gear was "linked up," in that the leads were then seriously affected, and generally in an adverse manner. Other faults with this type are wire drawing of the steam just at the moment of cut off, owing to the valve being moved slowly at that time, and a change effected in the compression or cushioning as well as in the "leads." Now with fixed single eccentric gears, such as named above, the lead remains a constant amount whether the gear is linked up or opened full out, and so they score well on this point. And the usual fitting of one of these is such as to give a quicker motion to the valve just about the time of cutting off, and hence the wire drawing loss is minimised.

There are, however, other points on which they do not excel, and these we shall mention in their proper places.

Hackworth's single eccentric gear may be taken as the earliest for our purpose, and consists essentially of one sheave placed with its line of throw exactly opposite to the line up through the crank. The eccentric rod went out in a practically horizontal direction, and its outer end was made with a long fork, and took a block which travelled up and down on an inclined plane, see fig. 41. This plane was capable of being turned by the reversing or weigh shaft until it occupied a corresponding angle in the other direction. From the eccentric rod attachment was made for what was called the valve rod

to come down from the end of valve spindle and be secured by means of a stout pin. The action resulting on the engine turning was as follows:—The sheave turning moved its end of the eccentric rod both vertically and horizontally, resulting in the block being at first pushed up (or down) the inclined plane and the valve rod also receiving a compound motion, the vertical component of which gave the movement to the valve. When the block was pushed or pulled down the plane it opened the top port to steam, and when pulled or pushed up the plane the bottom port became opened to steam. This gear gives a very good distribution of steam both to top and bottom ports, and the lead



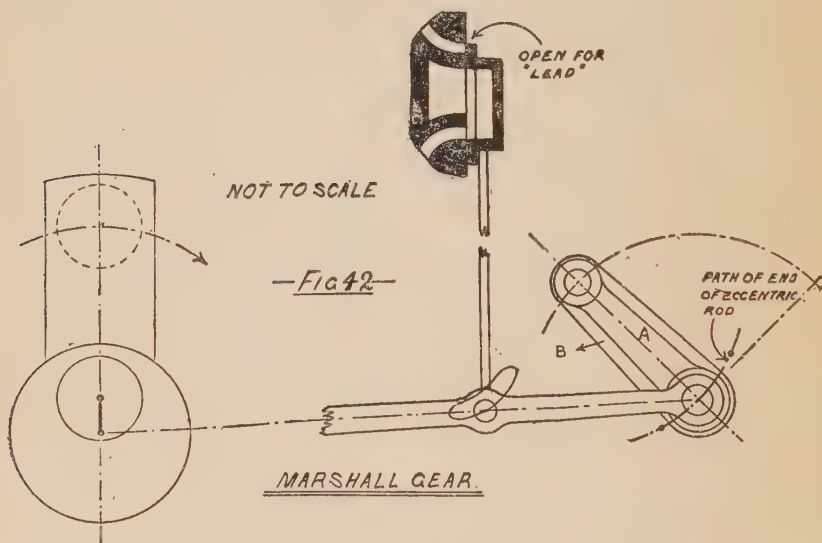
remains constant for all grades of variation in cut off. A gear the writer experimented with gave the following:—2.28 inches maximum movement upwards, *i.e.*, to open bottom port, and 1.8 inches maximum downwards, thus bringing out another rather valuable feature, *viz.*, the bottom port is opened more than the top one, and with an inverted engine this is just as we would like it, in order that a little more steam can be sent in underneath the piston so as to combat the weight of the reciprocating masses, which naturally tend to go to the bottom.

The following table gives results obtained from the experimental gear installed in the writer's laboratory, and though, of course, differences would be observed between this and a gear such as could be fitted

to a steamer, yet it shows clearly the good features of the Hackworth valve motion:—

MAXIMUM STEAM PORT OPENING.			
TOP PORT.		BOTTOM PORT.	
Angle.	Inches.	Angle.	Inches.
42°	1'28	42°	1'62
36°	'98	36°	1'2
30°	'7	30°	'83
24°	'44	24°	'56
18°	'28	18°	'36

The "angle" referred to here is that made by the inclined plane with the horizontal, and the diminishing angles show how linking up is effected by bringing the plane more and more to the horizontal.



Following closely the lines of the Hackworth gear is that known as Marshall's or Bremmes', which, whilst retaining the same principle, has one point or detail of difference, and that is the outer end of eccentric rod is taken by the lower end of a link *A*, which swings in an arc of a circle as seen from fig. 42. Now this results in a large difference so far as regards steam distribution, because as the eccentric rod moves,

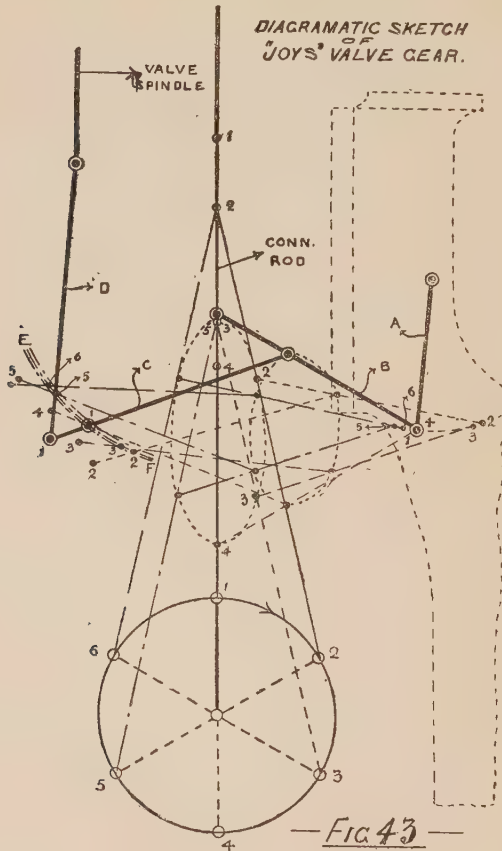
pushing and pulling upon the swinging link, it rises higher than what it falls. Let us show this point clearer, and to do which we must explain that in all valve gears a certain point like the centre of some pin exists, wherein a hole can be drilled and pencil inserted, and by holding against this a board with piece of paper on it we can obtain a graphical record of the motion of that point. The closed figure so drawn always comes out more or less of an elliptical shape, and the more nearly this approaches the *true* ellipse the better is the steam distribution. Further, we shall find if we fix the board in its first position and proceed to link up the gear that successive diagrams can be drawn, one for each grade of cut off, and all lying one over the other on the paper, and by careful examination of these we can see whether the leads remain constant and how the distribution is affected by our linking up. Dealing with the valve ellipses from single sheave gears we find they all cross each other at two points, and this tells us the leads remain the same throughout. And this latter point is proved also by an inspection of fig. 41, of the Hackworth gear, as there we see that the eccentric rod is of such a length as to be on the exact centre of the shaft upon which the plane turns when crank on top centre, and so however much this plane changes its angular position, there is *no* movement communicated thereby to the eccentric rod, and hence no change in the position of the slide valve due to us linking up. In figs. 41 and 42 are indicated what these valve ellipses are like, and whilst their actual shape varies with every different proportion of eccentric rod, angle of plane, length of swinging arm, angle of same, etc., yet the form always remains much as shown. One point to note with gears such as have been described is that they reverse very much easier than does a Stephenson link motion, as with this latter there are sometimes exceptionally severe stresses brought upon the valve spindle when the link is run over.

One or two features of the single eccentric gear in which they are not quite so good as the ordinary two sheave arrangement is that they cannot have their one eccentric moved round upon the shaft, and in some positions of the swinging arm of the Marshall gear there is a heavy stress brought upon the teeth of the reversing quadrant, which teeth are the only means whereby the reversing shaft is prevented from turning as it likes. With care in the design of the gear, however, this latter trouble is greatly minimised nowadays. Alteration in the travel of the valve can only be effected by either lining up or lining down the valve rod, or by altering the angle at which either the inclined plane of the Hackworth or the inclined arm of the Marshall gear lies, as the

more nearly this is brought to what may be called its mid position the more will the valve behave as it did with the Stephenson gear when the link was "middled."

Summing up the advantages of these gears they are: constant lead when linking up, fewer working parts, ease of reversal, a more open set of engines, less wire drawing of steam, and a quicker cut off.

With the Joy valve gear, as shown diagrammatically in fig. 43, we



see at once that no sheave is used at all, the valve receiving its motion from the combined effect of the travel of connecting rod and the movement of a fulcrum block in a curved path. The essential parts of this gear are the links *A*, *B*, *C*, and *D*, together with the curved path

EF, and the actions are as follows:—Directly the connecting rod moves as the crank leaves position 1, link *B* gets pushed along, and, its outer end being attached to the swinging or anchor link *A*, this moves in an arc of a circle. As soon as link *B* moves, it carries with it link *C*, which is really a rocking lever, being pivoted on a block which can slide along the curved path *EF*, and the outer end of this link *C* carries what may be called an intermediate link *D*, whose upper end is connected to the valve spindle. Then as link *C* is pulled along it moves its fulcrum block down *EF*, until by the time the crank has got to position 2 the other links and pin joints are also in the positions marked 2, 2.....2, in fig. 43, and the vertical distance between the two placings of the outer end of this rocking lever shows how much the valve has come down to admit steam at the top port. The crank in continuing its travel round to 3 brings all the different links, etc., also into positions shown by 3.....3, whence we see the valve has started on the return journey, and the fulcrum block is being pushed up the path *EF*, and so on for each new position of the crank until 4 is reached, when notice the block is back in the same place it had when the crank was on top centre. But now the outer end of link (or lever) *C* is *raised*, and this means the valve has the bottom port now open for lead, as the distance from fulcrum on the block to the outer end of *C*, where *D* attaches to, is such as will give a travel to the valve of (lap + lead), and then the movement bodily of the fulcrum block along *EF* gives the necessary steam port opening to admit the working steam.

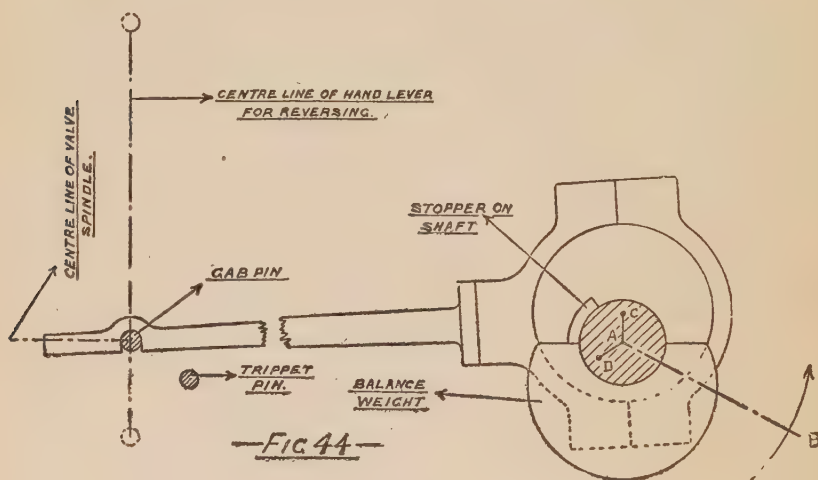
In the position of *EF* shown in fig. 43 the gear works in the "ahead" position, and to reverse it is only necessary to turn this until it occupies a position exactly opposite to that shown, and our reason for not drawing in this "astern" position was to avoid undue confusion amongst so many lines.

Fig. 43 well repays careful study, as it is interesting to trace out all the different positions that such points as the attachment of link *B* to the connecting rod, and of the rocking lever *C* to link *B*, make during one revolution of the engine; and we have indicated the shape the figures drawn through the successive points give, from which it will be seen they are of the elliptical form.

This gear is very easy of reversal, but unfortunately it is often very difficult to allow for the swinging link *A*, as being attached to the column sometimes leads to what can only be called a *peculiar* design!

Turning now to the loose eccentric gear used largely even at the present day in such as tug and ferry boats, we see the general arrange-

ment from fig. 44. One sheave only is used, and this is *not* keyed in a fixed position on the shaft, but is free to move upon this latter within predetermined limits. To balance this sheave there is fitted the special weight shown, and which also is available as the means of driving the eccentric. The dotted line *AB* shows centre of crank, and lines *AC* and *AD* the two lines of throw, but remember only one sheave is fitted. On the shaft is firmly secured a stopper, which can butt against either face of the balance weight, and is for the purpose of



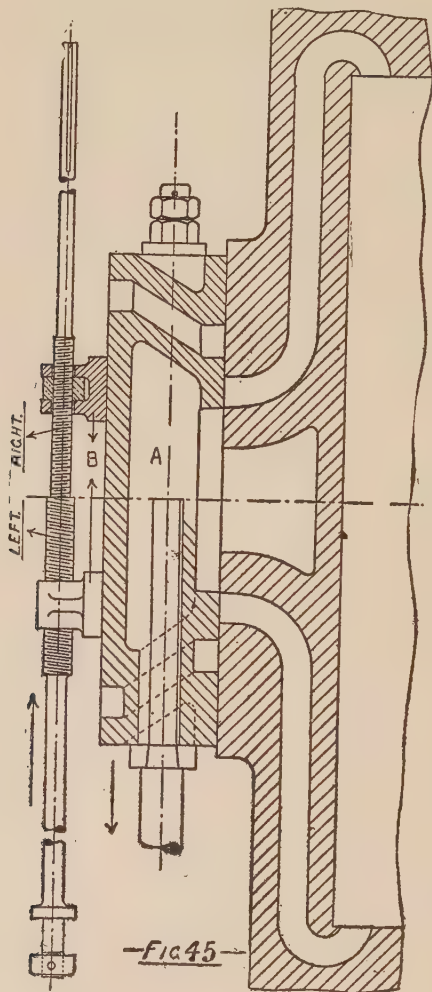
driving this latter together with eccentric, at the correct angular distance ahead of the crank. What happens when we wish to reverse is this. By means of the "trippet" pin, which is usually controlled by a foot pedal and rods operated by the engineer in charge, this pin is brought against the eccentric rod, and *lifts* it clear of the "gab" pin, which is upon the end of valve spindle, resulting immediately in the engine ceasing to control its slide valve. Then by means of a long hand lever, shown by its dotted centre line in fig. 44, and which lever is attached also to the gab pin, the engineer can, by either pulling or pushing at the upper end of this lever, move the gab pin, valve spindle, and valve into its mid position, whence the engine stops, and then as he sees whereabouts the crank has got to he knows which way to move his hand lever so as to bring the valve into position to admit steam on the *opposite* side of the piston, and so work the engine in the reverse direction. He follows the movement of the crank with his hand lever,

and when he has given the engine about $\frac{1}{2}$ or $\frac{3}{4}$ of a revolution by this means the crank has turned (and the shaft of course as well) sufficiently to bring the stopper on it up against the *other* face of the balance weight, which is, therefore, compelled to move itself and the sheave also. He then knocks away with his foot the trippet pedal, which has been held down by a little lug upon it engaging with a plate on the platform floor, and so the trippet pin falls down once more, allowing the eccentric rod end to come and rest for the time being upon the gab pin, and in a very short while, as the engine works obtaining its steam from the engineer's movement of his hand lever, the gab slot in the end falls upon the gab pin, and the slide valve becomes once more automatically worked by the engine, but which latter is now travelling in the reverse direction. In practice the act of reversal is very quickly carried out, and as we have immediate control over the engine directly the eccentric rod is off the gab pin, we are enabled to handle the engines very smartly, and this renders the loose eccentric gear most valuable for all classes of river boats. Whilst the crank is moving under the influence of the steam admitted by movement of the hand lever the sheave, being free and balanced, remains where it was until such time as the shaft stopper comes up against the opposite face of the balance weight, and when designing a gear such as this we must so proportion the stopper as to retain enough strength to drive the sheave against all valve friction, etc., and at the same time lay out the faces of the weight as to ensure the correct angle of advance being given the sheave.

With certain designs of the loose eccentric motion the sheave is arranged to *follow* the crank at 90 degrees less the angle for lap and lead, and this is useful if the valve itself is worked through a rocking lever, as then for the valve to move in one direction the eccentric rod has to move in the opposite, and so the sheave requires setting oppositely also.

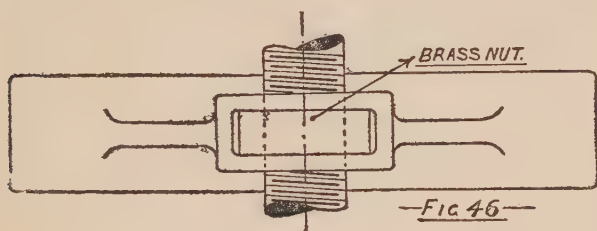
Turning now to a valve designed to give a sharp cut off of steam, somewhere say about $\frac{1}{10}$ stroke, if such be required, we confine ourselves to one type, viz., that known as the Meyer, also after its inventor. Fig. 45 shows it in position on the back of the main valve *A*. This latter is made in a different form to an ordinary flat slide in order that two passage ways, one at the top and the other at the bottom, may be formed from front to back. These passages constitute the steam inlets to the cylinder ports, and working upon the machined back of the main valve is the expansion valve proper, consisting of two plain cast iron bars *B*, having in their centre a portion cast with a

recess into which fits a gun metal or brass nut (see fig. 46), having for one bar a right handed thread, but for the other one a *left* hand screw is made, the functions of which we shall see about very shortly. The



expansion valve is driven by its own separate eccentric upon the shaft, and placed with its line of throw exactly opposite to the crank, and so leads that by 180 degrees when the engine goes either ahead or astern. Thus the valve does just as well for one way as the other.

The function of the expansion slide being only to cut off the steam, this it does as follows:—As the sheave actuating the main valve moves this, say, downwards, the sheave for the expansion valve is moving that upwards, hence the steam edge of the top bar soon meets the upper edge of the top steam passage, and thus the steam is entirely shut off. This point is independent of wherever the passage may be in respect to the cylinder port, as it must be noted that the cut off is with these valves effected on the back of the main slide and not the front as usual.

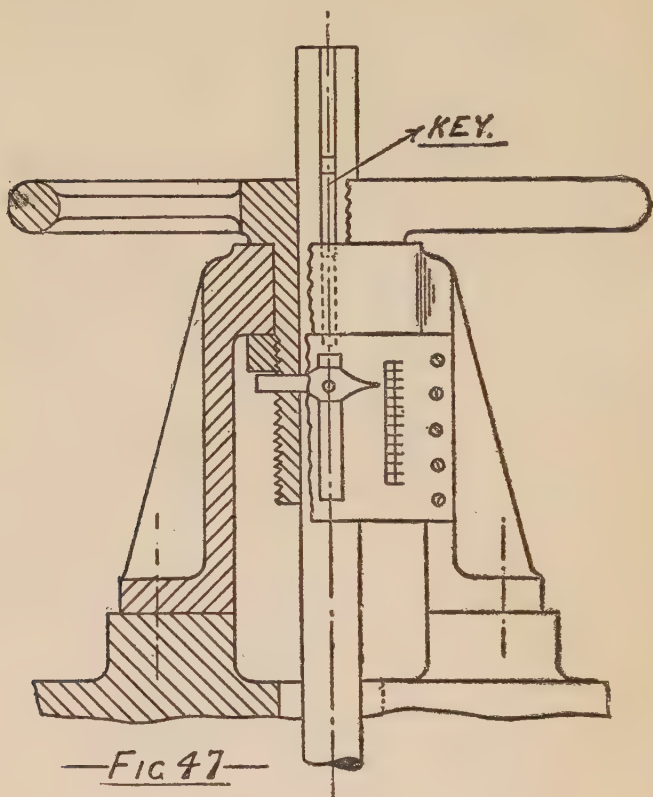


—Fig 46—

We saw in an earlier chapter that steam lap caused cushioning, and as this is just as necessary now as ever, we give the main valve such an amount of steam lap as will produce whatever cushioning we think the engine needs, but this steam lap will not be anything like so great as would be required for the sharp cut off of the entering steam. The travel, therefore, of the main valve is less, and the angular distance its eccentric is set with on the shaft is also less than for an ordinary valve.

To make the expansion valve as useful or handy as possible its spindle is made long enough to go up through a stuffing box fitted at the top of the steam chest as well as the one below it. On the top of this projecting spindle end is fitted a wheel handle, which has a sleeve attached, and which turns freely in a bridge piece surmounting. See fig. 47 (stuffing box omitted). The spindle has a long key or feather way cut down it, and in the above sleeve is fitted firmly a key, which is a good *sliding* fit in the spindle keyway, so that as this spindle rises and falls it moves to and fro through the wheel. The lower part of the spindle is attached to its own eccentric rod in such a manner as to enable the spindle to be revolved by means of the above wheel and key, and by so turning we are enabled to move the two bars *B* (fig. 45) either nearer to each other or further apart. This is most useful, as when they are run further out the cut off becomes sharpened, because the steam edge of bar *sooner* meets the upper edge of steam passage in valve, and as this alteration can be effected whilst the engine is running

we can adjust the cut off and consequent power and performance of the engine to a nicety.



—FIG 47—

It will be noticed from fig. 47 that a scale of cut offs is provided, against which a pointer controlled by the turning of the wheel and its screwed sleeve shows the particular grade of cut off we may have set the bars for, and when altering cut offs, a little care should be exercised to see that the *sharpest* allowance is not exceeded, as sometimes it is possible to get the expansion bars still further apart, and it may be this will result in a re-admission of live steam over the inner edges, and should this occur the engine runs erratically or in a series of jerks.

Expansion valves are usually fitted on the high pressure engine, and used to regulate the power and revolutions of the whole engine, and are of no use for any matter of balancing or equalising the horse-power between the different cylinders. To enable independent linking up to

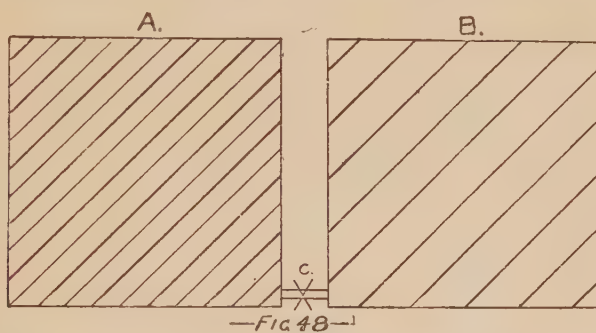
be carried out, *i.e.*, to alter individually the point of cut off on I.P. or L.P. engines, their reversing arm attached to the weigh bar shaft has a slot in its end of some inches length, and working in this slot, and controlled by a screw, there moves a block, carrying pins upon which the ends of the two drag links are attached. When the reversing arm is brought over by turning the weigh shaft all these arms of the three engines come over together, and so if one of the cylinders requires its valve to have the cut off altered we do it by moving the block in the slotted end by means of the screw, and so leave the other engines unaltered.

CHAPTER IX

THIS relates to the importance of balancing the indicated horse-power between each cylinder of a compound engine in order that the shaft may not be subjected to heavy and undue stresses set up by one cylinder trying to drive the others. Thus suppose a triple expansion engine of powers as follows:—H.P. cylinder doing 500, I.P. doing 400, and the L.P. only giving out 360 horse-power, and the usual engine driven pumps being worked by this latter cylinder. The total horse-power is 1260, and it is most important to remember that this total must *not be altered* to any appreciable extent by any change we may bring about in the individual cylinders. What we want is to take some of the large power off the H.P. and distribute it between the I.P. and L.P. cylinders, giving the latter one perhaps about 40 horse-power more than either of the others on account of its having the pumps to drive, and the power these need is taken off at the engine crosshead, and so never comes upon the crank shaft. If we let the engine remain as at first stated above the coupling bolts between the H.P. and I.P. shafts will be *unduly* stressed, because the H.P. engine is carrying the I.P. round with it, *i.e.*, driving upon it, in addition to the normal transferring of its own (the H.P.) horse-power thereto. At the outset of this subject we may say that no absolute rule can be laid down as to exactly what to do and to what extent in order to achieve a desired result. The whole business of equalising horse-power is largely one of experiment and careful noting of results from this, indicator cards being frequently taken and studied, also the horse-powers ascertained from these. Further, the key to the subject is “what space are we to allow exhaust steam of one cylinder to pass into,” as by this we alter its final pressure, and, noting as a last caution that the high pressure slide valve is *never* altered, we can commence investigating.

Imagine fig. 48 to represent two boxes, each of 1 cubic foot capacity, made steamtight and fitted with a tube and cock connecting one to the other. The volume of this tube we neglect in what follows:—Let *A*

be filled with steam of 160 lbs. gross pressure and represented by *all* the diagonal lines shown. The steam supply to *A* having been shut off, let us open the cock *c* and so allow the steam of *A* to fill *B* as well. We should not think that now the press. would be 160 lbs. nor yet is it;

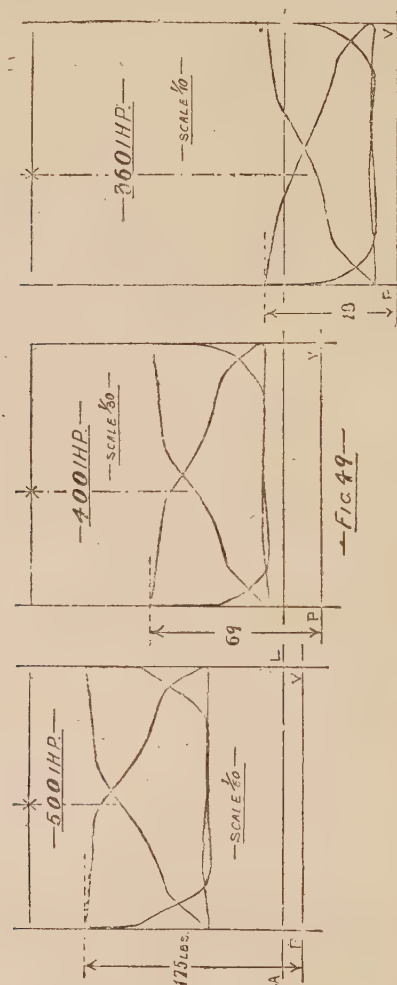


instead of this it is reduced to 80 lbs., or half, because the total volume of *A + B* is *double* that which was filled at first, viz., *A* only, and observe that this steam of 80 lbs. is at this pressure both in *A* as well as in *B*, and so from $Pv = \text{constant}$ we have at first 160×1 as the product of pressure by volume, and under the second conditions 80×2 or also 160. In fig. 48 by having half the number of lines in *B* we intend to convey graphically the fact that the original amount of steam is now occupying twice the volume. The reverse of what has just been written holds good equally, viz., that if a given volume be filled with steam of a certain pressure, and then that volume be reduced to half, the pressure within this lesser space will (bar losses) be *double* that at first. Or put another way, it is that if a definite amount of steam has to be sent into a volume which at one time is less than at another, the smaller the volume becomes the higher the pressure when all the given quantity of steam (say, exhaust from H.P. engine) has been got in. On the principle, as explained by the foregoing, lies the solution of all matters of power balancing.

Fig. 49 gives a set of three indicator cards showing the horse-powers we gave at first, and fig. 50 shows these as altered to bring the powers more into a balance. The differences are at once apparent on comparing the two figures. The points marked *X* on the stroke line represent the cut offs for each cylinder, and it will be seen that this is the chief *point* which is altered and in its turn affects the initial and back pressures on I.P., L.P. and H.P. cylinders.

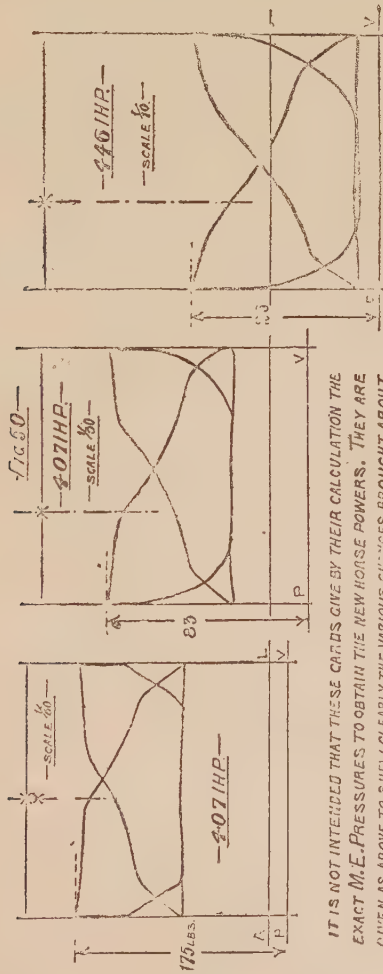
Now, to explain in greater detail, we must always take the cylinders

in pairs, thus, H.P. and I.P. or I.P. and L.P., never considering them singly. Adopting this we have the H.P. of 500 horse-power and the I.P. of only 400; evidently a balance will be got if each is made of 450 horse-power, and now how to get this? If we can do anything that



will compel the back pressure line of the H.P. cards to rise, then its area and subsequent power will be reduced, and whilst this exhaust or back pressure steam of the H.P. cylinder is the initial pressure steam (bar loss) for the I.P. engine, it is clear that if the back press. rises so

does the entering steam of the I.P., and so its steam line is drawn higher and the area of these cards in consequence becomes greater, hence the I.P. horse-power is more and so some comes off the H.P. and goes on to the following engine. This is what is wanted; but how

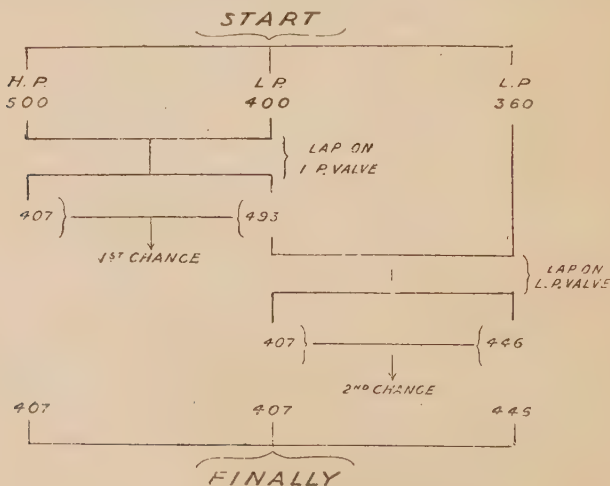


IT IS NOT INTENDED THAT THESE CARDS GIVE BY THEIR CALCULATION THE EXACT M.E. PRESSURES TO OBTAIN THE NEW HORSE POWERS. THEY ARE GIVEN AS ABOVE TO SHOW CLEARLY THE VARIOUS CHANGES BROUGHT ABOUT.

is the slide valve going to accomplish it? Very simply, by merely putting additional steam lap on I.P. and advancing the eccentric to regain the old lead or else link up the I.P. gear. Either of these alterations makes the cut off in the I.P. engine occur sooner, which means less

inches of the piston stroke have been done when the valve closes the port against the entering steam, and so the volume open to receive the exhaust from the H.P. is *less* than before and so the pressure in this reduced volume must be higher, by what we explained in regard to fig. 48. In the case we have given at first, it will not be sufficient to get the H.P. and I.P. cylinders each doing 450 horse-power, because we have to consider the L.P. which is only doing 360 I.H.P., and we wish her to put out 446 or to express the new horse-powers of each, they will be 407 H.P., 407 I.P., and 446 L.P., or a new total of 1260 horse-power, the same as what we had at first. We see therefore directly that to alter the I.P. valve to make the H.P. engine yield 450 H.P. will not do, as that cylinder must only exert 407 H.P., and so whatever we did in the way of lapping the I.P. slide, etc., must be carried a little further and pull down yet more the power of the H.P. engine, and at the same time increase the I.P. Note that for the moment we purposely upset the balance between these two engines, but by now having the I.P. a good bit higher than we want, viz., 493 horse-power, that the pair of cylinders now being worked upon are the I.P. 493 and L.P. of 360, and these must be adjusted to 407 I.P. and 446 L.P., to do which the L.P. slide valve must be lapped and eccentric advanced, or the gear linked up to give the requisite sharpening in the cut off, as by so doing the back press. of the I.P. rises and so pulls down its power, whilst the forward steam for the L.P. becomes of a higher pressure and so its horse-power gets more.

Summarising the foregoing into a table we can follow the different steps more clearly and concisely thus :—



Remarking upon the foregoing we would draw attention to the following facts. The engine that requires an *increase* in her power is the one whose valve needs its steam lap *increasing* or has to be linked up, whichever we think will achieve the end best. On this detail one cannot say exactly what to do, as some engines will stand a great deal of linking up, whilst others begin to knock almost as soon as the link is touched. With these latter practically the only thing is to piece the valve and advance the sheave. Bear in mind also that the high pressure slide is *never* to be touched. The reverse of the foregoing paragraph holds good as well, viz., the engine which needs her power *reducing* requires her valve having its steam lap reduced in order to cut off later, or else, if she has been running linked up, to put her back to full gear, but again remember the H.P. valve must not be touched.

It will be found sometimes that an engine actually revolves in the smoothest manner when the powers between the different cylinders as transmitted down to the crank shaft are not all equal, and the exact reason for this is not understood, but for want of any better idea we may attribute it to the individual nature of the engine, as all those who have had experience in the running and handling of such well know how one gets to learn an engine's peculiarities or its nature. When engaged balancing powers, therefore, always be guided by the performance of the engine itself after each alteration has been made, and also observe what effect is produced upon the coal consumption. When this has been brought to its minimum leave well alone and do not touch the engine again. (See also end of this Chapter.)

We now give an example where the powers are again out of balance, but in a different sequence. For example, the H.P. is doing 400 horsepower, the I.P. 480, and the L.P. 550, with no pumps on either engine, as whilst the bilge pumps are certainly on one or other cylinder still for our purpose here we can neglect them. Now at first sight one might be tempted to open out the H.P. or take off some of the steam lap of this slide valve, and so increase the power of that engine, as it is clearly too low, but would the H.P. really be increased by opening out its gear or taking steam lap off, because we must remember that it is not a *single* cylinder only which is in question, but one of a set of three forming a triple compound, and as the power of a previous engine is affected by the volume of the immediately following one, into which it can pour its exhaust, the power of our H.P. here might remain much as it was at first. When we open her out and carry steam further the steam line is longer and expansion curve less, with the pressure at opening to the exhaust greater also, and this means the steam going to

the next cylinder, having the same volume to receive it as before, is greater and of a higher pressure, and so the back pressure is more on the H.P., and it is possible that this counteracts the gain due to larger area of diagram under the steam and expansion lines. However, it is the custom usually to look upon opening out the H.P. as increasing its power, whilst closing in of its gear decreases it, but the remarks in the preceding paragraph and also those a little later on in this chapter should be remembered.

If it were only a question of one cylinder on its own then at once linking up or lapping would lower, whilst opening out or taking off lap, etc., would increase its horse-power. Reverting, then, to the first part above, as the H.P. exhausts at a higher pressure and more of it means more steam, and so the others would get more and their power would go up, and finally the balance (?) would be just as bad as at first. No, the H.P. valve must not be touched. Then what are we to do? The total power is $400 + 480 + 550 = 1430$, and as before this must not be altered, at least not to any serious extent. Neglecting the pumps we then need $\frac{1430}{3} = 476.67$ or, say, 477 horse-power from each engine, hence the H.P. must *rise* from 400 to 477, whilst the I.P. goes *down* from 480 to 477 and the L.P. goes *down* from 550 to 477. If we make the cut off in the I.P. later we give a greater volume for the exhaust of the H.P. to go into, and so the final pressure in this larger volume will be *less* and both back pressure on H.P. and forward pressure on I.P. are lower. Thus the area of the H.P. cards becomes greater with a corresponding increase of power of that engine, and the area of the I.P. ones becomes less, and so its horse-power is decreased. Now observe that there is a considerable increase in the power of the H.P. engine, whilst the I.P. goes down only a trifle and we shall find that we have to alter the I.P. valve such an amount in order to get the big rise on the H.P. power, that for the moment the power of the I.P. engine is below what we require. This is cured in the next step of balancing the L.P. together with the I.P., thus the I.P. might be *now* only giving perhaps 403, with the L.P. still 550, so it wants 74 more horse-power, that must be got from the L.P., leaving this engine with $550 - 74 = 476$ horse-power, quite near enough to the amount we fixed at first. To effect this change we must take lap off the L.P. valve and back the eccentric, or else put her into full gear again should she have been running linked up. This results in a greater volume of the L.P. cylinder being open to receive the exhaust of the I.P. and when this greater volume becomes filled, the pressure in it is less, hence the

back pressure of the I.P. is lower and the forward pressure on the L.P. is also lower. The area of I.P. cards is made greater by this lowering of its back pressure lines and so its horse-power is brought up again and by a little careful trial and observation of results of each alteration of the slides, we shall be able to effect the desired balancing.

The writer of this work was engaged at one time upon a case of equalisation of horse-power between the cylinders of a triple compound, and when the best running point of the engine had been obtained, in conjunction with as near a true balance of horse-power as possible, the following marked results were found, the speed of the ship had *increased* by fully $\frac{1}{2}$ knot per hour, whilst the coals had *decreased* by a matter of 2 tons per day, thus showing a most excellent result for all the time spent in trials, because actually several months elapsed from the time of taking the matter in hand until the above were obtained.

A few words as to the various effects produced by an independent linking up of the individual engines will form a fitting termination to this chapter, and should be read in conjunction with the remarks upon page 87, dealing with opening out the H.P. engine. Suppose the H.P. engine to be linked up only, then a smaller quantity of steam passes in, due to the cut off being sharper, and as the expansions are now greater the pressure at opening to exhaust will be less, even though this latter occurs also a little earlier, say, than in full gear. This means that the back pressure on the H.P. is less, and so the area of its card will remain much about the same as before, etc., therefore its horse-power is not sensibly altered. But this lower back pressure is also a lower forward press. for the following cylinder, and hence its horse-power comes down, even though its own back pressure will be lower, but whereas the H.P. card showed a loss of work *only* on the expansion curve, the I.P. card shows a diminution on its steam line as well as expansion curve, and most likely the lowering of its back pressure line will not compensate for this. These latter remarks apply also to the L.P. cylinder.

If now the I.P. engine only is linked up, it makes the back pressure on the H.P. higher, and its power less; the I.P.'s own power goes up, and the L.P. does not appreciably have its horse-power affected because whilst the entering steam of the I.P. is higher, yet it goes through more expansions and the exhaust and back pressures may therefore be less and as this steam is the forward pressure for the L.P., we find the admission line of this latter engine is lower and as it has not yet been linked up, the exhaust will also be lower in direct proportion to the lower entrance pressure. This means that the steam being at a

lower temperature when it goes down to the condenser, the working vacuum will be better and this may compensate for the lowering of the admission line.

Grouping together the gist of the previous pages we obtain the following :—

Linking up on	
H.P. only }	Makes total horse-power less, but its own is not appreciably affected. I.P. power is less, whilst the L.P. may or may not be lowered.
I.P. only }	Raises back press. of H.P. and its own initial press. is more, therefore horse-power of H.P. is less, whilst I.P. gains. L.P. power is usually not altered. Total horse-power remains the same.
L.P. only }	Leaves the H.P. as it was. Makes the back pressure on I.P. higher and its own initial pressure more, therefore L.P. power is greater, whilst I.P. loses. Total horse-power remains the same.

One must remember that in everyday practice things have to be modified a little, over and beyond what strict theory would dictate, but all we have written here should be well grounded in one's mind, as it forms a good foundation for any alterations of power.

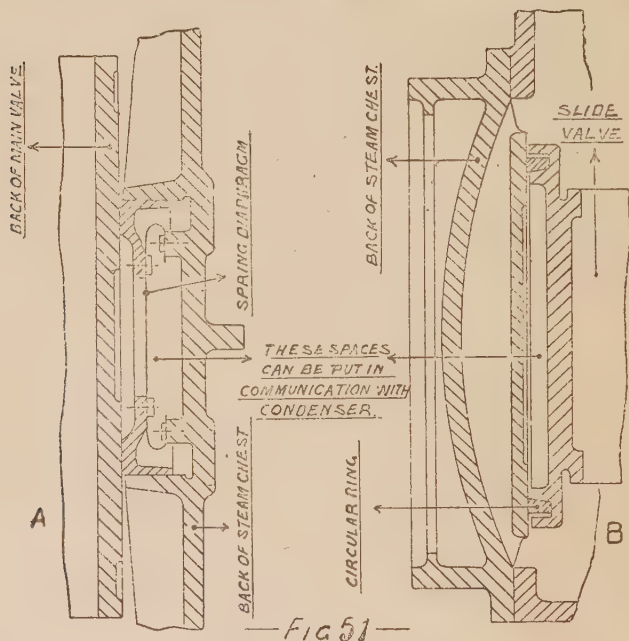
To be able to carry out this adjustment or balancing in the best manner it is desirable to have independent linking up fitted to each engine, as described on page 81, and even with this it is occasionally necessary to insert liners between the ends of the brasses and the collars on the drag links, in order to increase the distance between the centre of the pin on block in reversing arm and the centre of the drag pin on reversing link, and so obtain a little more "linking up."

CHAPTER X

DEALING with the friction of slide valves, and some of the means taken to minimise the loss from this, we have first to recognise that it is only flat valves which cause heavy friction to exist between their own face and that of the cylinder.

When explaining the "piston" form of slide we drew attention to their greatly reduced friction and, in fact, this latter only exists when spring packing rings are used to ensure steam tightness in the respective upper and lower "pistons." Dealing then only with flat valves, we should firstly like to ascertain how much the effort actually is that is necessary to move the valve to and fro, but to get anything like an accurate answer to this is impossible. That it is at times great we know only too well from such as eccentrics and straps running hot, and in some cases the white metal melts out. But with reasonable care and proper lubrication we need not fear trouble on this account, always provided the surfaces of strap and eccentric have been *correctly designed* for a fair estimate of the force required to be put out to overcome not only the valve friction but also that of the spindle in its stuffing box. If we trace the actions going on for one stroke we shall see how the force pressing slide against the cylinder face varies. Assume the valve has the top port open for lead and steam passing into cylinder. So long as this goes on the upper part of valve may be considered in equilibrium, but shortly cut off occurs and the steam inside the cylinder expands and re-acts back through steam passage on to the upper face of the valve, and so this part has on its back, steam of the pressure in the chest forcing it on to the cylinder face, whilst the front part has a constantly reducing pressure acting *against* this. Presently the top port opens to exhaust and now steam of a pressure determined by the amount of expansion in the cylinder fills the exhaust cavity, and this acts also in opposition to the steam pressure on the back of valve and at the same time the bottom port is open for the admission of steam, and so the lower face is for the time in equilibrium just as was formerly, the upper one. Proceeding, the valve comes down to cut off the exhaust at top port and so

cause cushioning of the steam left in the upper part of the cylinder, and cut off has already occurred at the bottom, so we have the upper face feeling the effect of a constantly increasing pressure which with a bad setting of the slide valve can easily become *more* than the pressure in the chest, and which tends to force it off the face. At the bottom a pressure also exists tending to force off the valve, but to a very much lesser extent, we therefore see that though the pressure in the chest *may*



—FIG 51—

remain constant (as a matter of fact it varies quite appreciably also) yet we are by no means certain just what number of lbs. to allow as an opposing factor tending to ease off the valve. Further, the condition of the surfaces affects the friction very much; thus if they are smooth and well lubricated it will be small; if smooth but dry it will be greater, and if rough and dry as well the friction co-efficient gets very high.

Instead of spending a long time in guessing at what to do, let us be content to make the best of a bad bargain and inquire as to what can be done to minimise the loss of work in having to drive slides against their frictional resistance.

Obviously a valve with a large area of back open to the steam pressure of the chest will require much more effort to drive it than a

small back one having the same pressure per square inch against it, and this gives a clue as to what is done to save work. Some of the back area of the valve is removed from the pressure of the live steam, and fig. 51 shows two methods of carrying this out by means of "relief rings." Method *A* shows a section through back of chest, relief ring, and main valve, the ring being a circular casting made either in cast iron, brass, or gunmetal, and working in a turned recess cast in the back of chest. To hold it against the back of valve and to allow for slight wear, a spring diaphragm plate made of a hard springy brass is secured by bolts to a projecting ring cast on the steam chest, and also to the relief ring itself, and there is formed by this an enclosed space which may if we desire, and usually is, put in communication with the condenser, so that a vacuum of a very good degree exists in this enclosed place and whatever steam pressure there may be on the similar area of valve, only on its inside surface, will be able to exert as many *more* lbs. effort as what there are lbs. of vacuum. Remember that speaking commonly all steam pressures are quoted as so much above the atmosphere, and in dealing with such as is now being written upon, the pressures should be given as gross. A few figures may help to get the foregoing well fixed in one's mind. Assume the valve an L.P. one whose area of back is say $4' \times 4'$, or 16 square feet, or 2304 square inches. Next suppose we assume an actual net pressure forcing this valve on to the cylinder face, of (7 lbs. + 15 lbs.) - 5 lbs., and which 5 lbs. is taken to allow for variations and rise due to cushioning and is not to be looked on as the actual back pressure due to the vacuum being carried in condenser, thus giving 17 lbs. per square inch, or a total of 39,168 lbs. coming on the back of valve. Of course this is not the effort which has to be overcome, as we must consider the friction coefficient, and taking this at about .2 we have $39,168 \times .2 = 7833.6$ or roughly 3.5 tons effort, truly an enormous amount.

Now, supposing there is fitted to the valve a relief ring of possibly 2 feet diam., or an area of 3.1416 square feet, and this amount is at once taken away from the 16 square feet we had at first, leaving us 12.85 square feet. Not only have we this reduction, but seeing there is now an area open to the vacuum, we have, when the valve moves into such positions as bring steam above the atmospheric pressure on to the opposite side of the valve to what the vacuum acts on, an augmentation of their pressures, and so they become as it were more energetic in endeavouring to ease off the valve; and to allow for this effect we may take the average opposing pressure at perhaps 7 lbs. instead of 5 lbs., giving us then only 15 lbs. pressing upon the *reduced* area of 12.85 square feet,

and so the total load is now only 27,756 lbs., and with the co-efficient of $\cdot 2$, we have 5551 $\cdot$ 2 lbs. now, showing a considerable reduction. Of course the above figures must by no means be taken as from an actual case, as you cannot obtain all the data one requires accurately, but they show very clearly the saving by using relief rings.

Seeing, then, how useful these are, it is a pity they are liable to a very serious fault, viz., leakage past the face of ring and planed up back of valve. Should a piece of dirt or the like get wedged between these it either keeps the ring permanently off the valve, or it scores a groove in one or other surface, and so the steam in the chest obtains access to the previously enclosed space, and so into the condenser, where it is turned into water without having given up any work in the engine.

The Board of Trade have two arithmetical questions bearing upon this from which the following has been arranged :—Suppose the total area of leakage between relief ring and back of valve to be, say, $\cdot 2$ square inch, gross pressure of steam in the chest 22 lbs. per square inch, required the loss in lbs. of steam for 24 hours by the above crack. The rule given by the Board says that in “every 70 seconds there are as many pounds weight of steam escaping as there is gross pressure upon the area of the crack.” Applying this here we have 24 hrs. $\times$ 60 m. $\times$ 60 secs. = 86,400 seconds in the given time of 24 hours; $86,400 \div 70 = 1234\cdot 3$, and neglecting the decimal this is 1234 “seventies,” in *each* of which there escapes a weight of $\cdot 2 \times 22 = 4\cdot 4$ lbs. of steam, or a total leakage of $1234 \times 4\cdot 4 = 5429\cdot 6$ lbs., and remember this goes on as long as the engine runs. If we take the evaporation to be at the rate of 8 lbs. of water per 1 lb. of coal, we shall obtain $5429\cdot 6 \div 8 = 678\cdot 7$ lbs. of coal lost every 24 hours, and so on a voyage of, say, three months’ duration (91 days) there will be a total loss of coal of $\frac{678\cdot 7 \times 91}{2240} = 27\cdot 5$ tons,

and if each ton cost, say, 11s. the actual monetary loss by this, at first sight insignificant, leak, is $\frac{27\cdot 5 \times 11}{20} = \text{£}15 \text{ 2s. 6d.}$ And this is not the

end of the losses or expense, because the leakage steam going to condenser means that in order to keep the same vacuum therein we shall have to send more circulating water through in a given time, and this means expenditure of more work, and even should we be content to let the vacuum drop back a little and so let us have hotter feed water and saving a trifle in the boilers, remember the reduced vacuum means less work obtained from the L.P. cylinder.

It should be quite clear by now that whilst relief rings are most valuable for easing the friction of slides, yet they require a good deal of

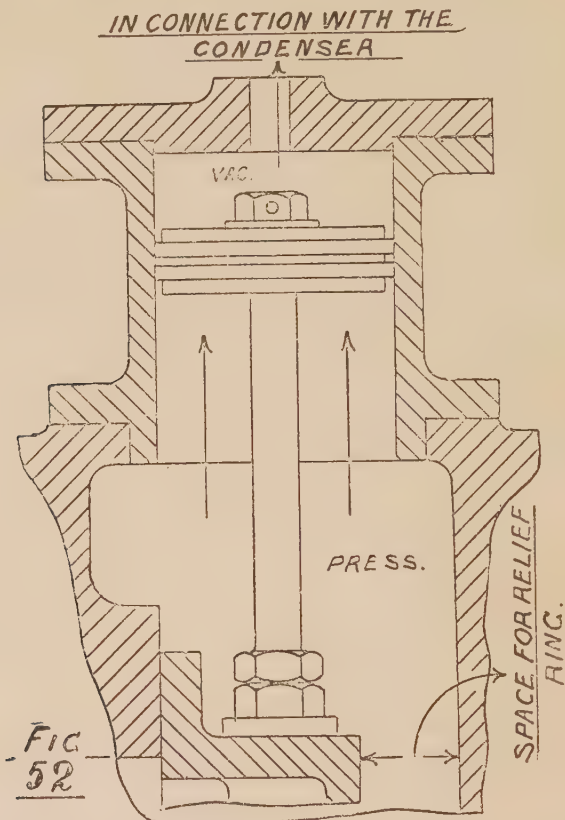
attention whenever the engines are opened out so as to make certain the surfaces are not scored, or if they are to take them out and reface truly.

In *B*, fig. 51, is shown another style of relief ring, and one of much larger diameter, and suitable for a valve of very small travel, thus still further reducing the face friction of slide on cylinder, and it consists of a coned ring, split at one place so that it may adjust itself as it works upon the coned shaped part of the back of valve, in order to keep itself against the machined portion of the steam chest. So that this split may not pour steam down to the condenser, there is fitted a tongue piece, and in fact the whole ring is just the same as a packing ring for an ordinary steam piston, with the exception that the inside of it is turned conical instead of parallel. These rings may be made either of brass, gunmetal, or cast iron, but this latter is really not suitable and only used on account of its cheapness.

In order to promote steam tightness between the relief ring and back of valve, they are in many designs held up to this latter by springs pressing upon the ring and capable of adjustment by studs passing through bosses in the back of steam chest. The slight extra friction produced by such is more than counterbalanced by the lesser risk of leakage, and further, where the ring is of the sliding type, shown in *A*, fig. 51, that portion which is actually moving in and out sometimes has grooves turned and into these soft packing is fitted to ensure a steam tight job.

As well as relief rings to ease the face friction, there are often fitted in the case of heavy L.P. valves what are called "balance cylinders," these in their simplest form being a small cylinder placed on top of the steam chest and containing a piston attached to the valve spindle continued up through what would have been the steam chest cover. On the underside of this piston, as seen from fig. 52, the steam at the pressure of that in the chest has access and so tends to force *up* the piston carrying valve spindle, slide, gear, etc., and so very largely helping to take off the weight of all this from the eccentric. We must however, be careful not to make the diam. of this cylinder too great, or else the upward effort will be so large that when the sheave has turned to move the valve and gear *downwards* it finds considerable opposition from the effort of the steam to keep them all up. What we must aim for is to have the area of piston such that when multiplied by the available gross pressure of the steam on its underside, that then the gear, etc., will be just "balanced," that is, has neither the tendency to fall nor yet go up, and then the eccentric has the minimum work to put out. The space between top of piston and cylinder cover is put into

connection with the condenser and so we can get more available work out of the steam, but this being the case, such balance pistons are liable to all the troubles of leaks that we wrote about when describing relief rings.



A more modern form of balance piston or cylinder is that known as Joy's "assistant cylinder," the piston of which is secured to a continuation of the spindle, but is of such a construction that not only is the valve, etc., helped in its rising, but means are provided whereby the rush to the ends of the stroke of the valve, spindle and gear, are checked by cushioned steam, thus making for a smoother working gear and also helping the parts to start their respective strokes. It is therefore essentially a cylinder and piston which counteract the inertia of the moving parts of the valve and motion, whereas the ordinary balance piston is merely a means of neutralising the deadweight of the parts.

CHAPTER XI

IN this we propose to deal with the application of slide valves to such as feed pumps, ballast pumps, and steering gears, noting individual peculiarities and showing how important it is that the slides perform their functions correctly for these auxiliary machines, just as it was in the case of the main engines.

Starting with feed pumps, we take the well-known and excellent type made by Messrs. G. & J. Weir, of Glasgow, and to which firm we tender our best thanks for so kindly placing illustrations and descriptive matter at our disposal. The figures given in regard to these "Weir" pumps are all as issued by the makers, and readers are advised to carefully study each one, as they are drawn in a nice clear manner and not hampered with a lot of unimportant details.

Whilst by rights we should confine ourselves only to a description of the slide valves, yet as this would leave matters more or less unsatisfactory, so far as the pump as a whole goes, we make no further apology for the following description:—

Fig. 52a shows a sectional view of a Weir pump used for boiler feeding, and which consists of an upper steam cylinder mounted directly over the pump cylinder at the bottom, the piston and pump rods, 10 and 21, being secured into the main crosshead, 8, the whole working straight up and down. The pump, it should be noted, is double acting. To operate the valves controlling the admission and exhaust of the working steam there is fitted a crosshead pin, 9, working the "valve gear levers," 5, which in their movement vertically bring the "ball crosshead," 7, against the collar and nuts on the "bottom spindle," 4, which, being pivoted at the "double joint," 2, connects to the "auxiliary valve spindle," 26, which carries the small or auxiliary valve whose functions are to admit steam to operate the "main valve" and also to cut off the entering steam. 1 shows the end of "steam slide valve chest," and 14-19 the pump valves, which are arranged in groups, there being generally one less in the discharge set than in the

suction in order that the discharge may be more regular and at a little higher pressure. The maximum lift is $\frac{1}{4}$ inch. The "pump bucket," 20, has in gunmetal lined pumps packing rings of ebonite fitted, but

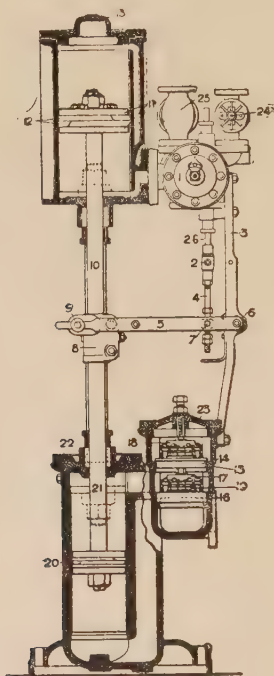


FIG. 52a.

Weir Feed Pump. Sectional View.

in cast iron lined pumps these are not used, and instead a cast iron ring is fitted, having water grooves turned in its periphery.

Details of the steam valve are as follows:—It consists of a main valve *A*, fig. 54, and auxiliary valve *B*, figs. 54 and 56. The main valve moves horizontally, being driven by steam admitted and exhausted to each end of itself alternately through the ports *E* and *F*, fig. 54, and in its travel to and fro it covers and uncovers the ports *C* and *D* leading to the bottom and top of the cylinder. This main valve works in two loose caps, seen in fig. 54, and which are held in position by the end covers and faces cast on the chest for that purpose. The auxiliary valve *B*, figs. 54-56, is the one actuated by the lever gear driven off the piston rod, and moves on a face formed upon the back of the main valve, which, it may be stated here, is circular at the ends

THE WEIR STEAM VALVE.

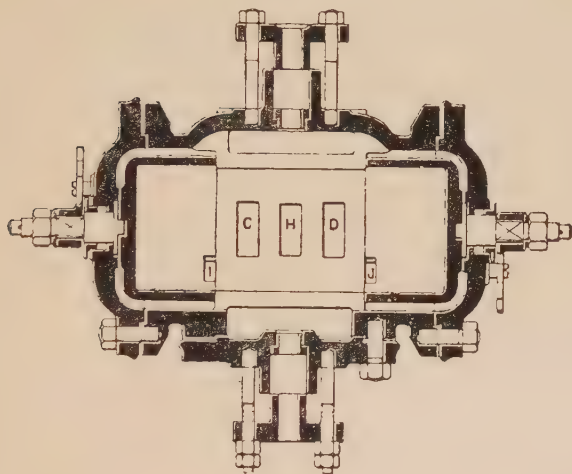


FIG. 53.

Sectional Front Elevation, showing Bye-pass Ports I and J.

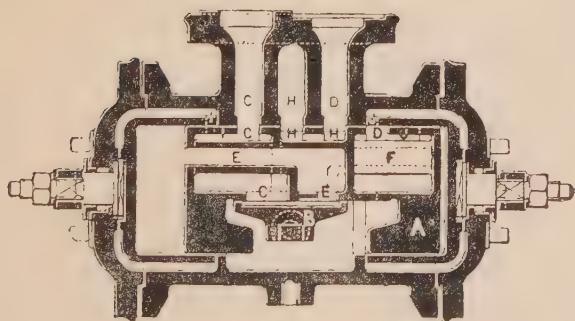


FIG. 54.

Sectional Plan.

but half round at its middle portion, as seen from fig. 58, thus the auxiliary valve moves at right angles to the main valve. By this arrangement there can be no dead centre, because the main valve *A*, can only be at one end of its horizontal stroke or the other. In figs. 53, 54, 55, 57, 58, *H* shows the exhaust port and passage. When the piston is at the bottom of its stroke the main valve is in its right hand position, with port *C* leading to the bottom of cylinder open for steam admission. This port remains open until the piston has reached half stroke, when auxiliary valve *B* begins to move upwards also, and at about $\frac{3}{4}$ stroke this latter valve has closed port *C* to steam, and so the rest of this up stroke is done by the expansion of the steam in the cylinder or by more steam admitted through the by-pass, which we will describe shortly. When the piston reaches the end of its stroke the auxiliary valve opens the exhaust port *E*, figs. 54 and 58, and by the exhaust cavity in this valve allows the steam which has been at the left hand end of main valve to escape, and the other end of this latter valve being then open to steam through the port *F*, fig. 58, the valve gets blown over, as it were, to the left by virtue of the steam getting between the right hand cap and end of valve, as seen in fig. 54. In its travel across it moves at first with a fairly high velocity, but presently there comes an instant when, owing to the shape of exhaust cavity in auxiliary valve and port *E* in the main valve, the exhaust from the left hand end of this is shut, and what steam then remains acts as a cushion, and ensures the main valve coming quietly to rest instead of striking the cap a severe blow. When the main valve has come to its new position, *i.e.*, the opposite to that shown in fig. 54, port *C* is now open to the exhaust *H*, and port *D* leading to the top of piston is open to admit the working steam for the down stroke of pump, and the same actions as just described all take place one after the other for this stroke.

We have stated that the steam is cut off at about $\frac{3}{4}$ stroke and by expanding completes this, but there are occasions on which the drop in pressure, etc., due to this expansion might be such as to leave insufficient to work the piston against the pressure in the pump cylinder, such as, for example, on starting, owing to the steam cylinder being cold, the steam kept in and expanding would suffer heavy condensation and large fall in pressure, so to overcome this it is necessary to admit a little more steam after the auxiliary valve has closed ports *C* or *D*. To do this a by-pass is arranged for each end of the cylinder, and which can be opened by hand, allowing steam to flow in. These are shown at *I* and *J*, figs. 53 and 57, and consist in cutting small ports in the caps at each end, fig. 53, and elongating the lower parts of the two

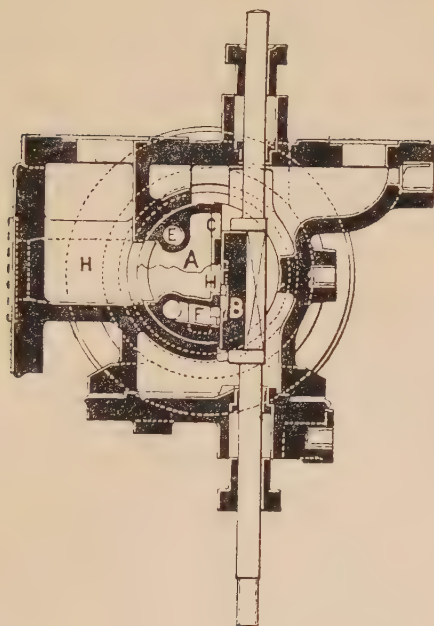


FIG. 55.
Sectional Side Elevation.

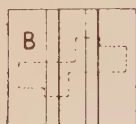


FIG. 56.
Auxiliary Valve.

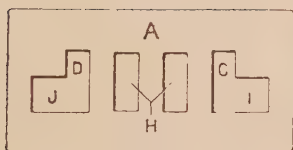


FIG. 57.
Main Valve Face.

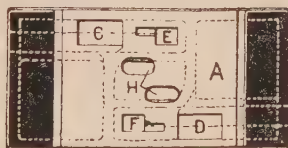


FIG. 58.
Main Valve Back-Face for
Auxiliary Valve.

ports *C* and *D*, fig. 57. As arrangements are made whereby the caps can be turned round from outside the valve chest, it is quite a simple matter to cover or uncover these by-passes as may be desired.

Messrs. Weir's later type of pump is fitted with what is called the "high pressure" steam valve, shown in fig. 59, and which is the type most suitable for the feeding of water tube boilers working at pressures of 250 lbs. per square inch and upwards. The principal difference between this valve and the one previously described lies in the by-pass fitting.

From fig. 59 it is seen that the caps are done away, the chest being made longer to form the cylinder at the ends, and at the top of the steam chest two parallel plugcocks, *K* and *L*, are fitted, one for each end of cylinder, and these act exactly as did the by-pass ports mentioned above. It is not necessary to leave these open after pump is started, as, if they were, the steam being admitted right to end of stroke would cause the pump to knock badly.

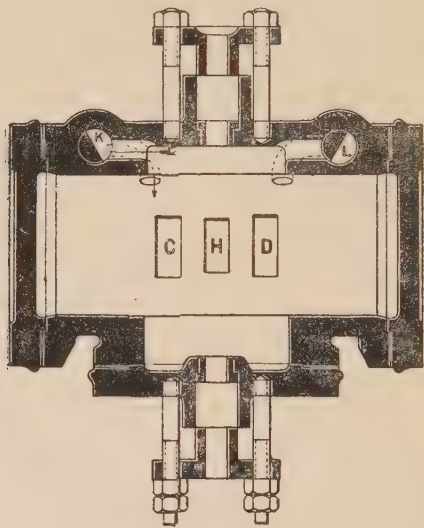


FIG. 59.

Whilst we are writing as though these pumps were only feed pumps this is not the case, as they are made by Messrs. Weir to act as oil fuel, forced lubrication, hotwell, latrine, drain tank, fire and bilge pumps, there being only modifications in the design to suit them for their individual duties, but the actions of the slide valves, which is what we are mostly concerned with here, remain the same.

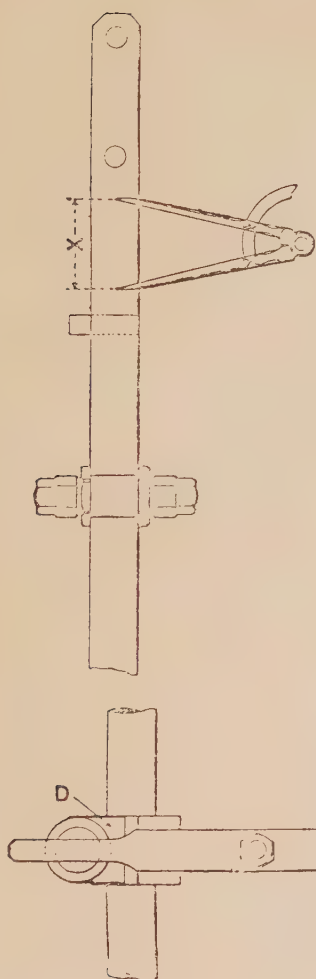


FIG. 60

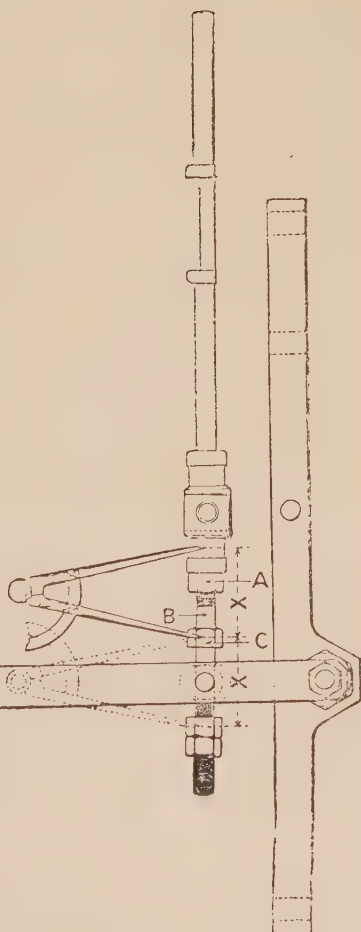


FIG. 61.

Method of Adjusting Length of Stroke.

Air pumps, both of the dry and wet types, are also fitted with these slide valves, and in certain classes of work Messrs. Weir are known as well for their makes of "Dual" and "Monotype" patterns of air pump as they are for feed pumps. In addition they also make a combined set of air and circulating pumps.

One more point of importance in connection with these excellent pumps is that of adjusting the length of stroke. Should it be found necessary to readjust this, set a pair of dividers to the distance marked *X*, fig. 60, which shows the front stay of pump, and upon this two centre pops are marked by the makers, then adjust the nuts on the bottom spindle *4*, fig. 52a, until the centre pops upon them and the collar *C* corresponds with amount *X*, see fig. 61, then lock the nuts hard before starting the pump. Also take the compasses, opened to the same amount, and make the distance between pops on collar *C* and on the "bottom spindle" itself, also equal to the amount *X*, fig. 61.

Another method of adjustment irrespective of these centre pops is to slacken back the nut *A*, fig. 61, and gradually screw up the bottom part of spindle *B* by means of the hexagon collar *C* till the piston strikes the cylinder cover. Measure the distance between the cylinder gland and top of crosshead *D* on the piston rod. Then screw spindle *B* downwards till this distance is $\frac{1}{2}$ inch more than before, that is such that as pump works the crosshead goes up $\frac{1}{2}$ inch short of what it did at first. This gives the necessary clearance for smooth working on the top, so repeat the process, only measuring from bottom of crosshead to pump gland, and thus set the bottom clearance. By further closing in of the nuts the stroke can be still further reduced if desired.

Sufficient has been written, we think, to show how these pumps work so far as their steam end is concerned, and we would strongly advise any engineer having these under his control to write to Messrs. Weir, Cathcart, Glasgow, and obtain a copy of their pamphlet relating to pumps if he has not already one, and we know that every assistance will be given him.

Fig. 62 shows the steam cylinder end of a 10-inch stroke Worthington pump, and its peculiarities are that two ports and passages are made at each end; also the nuts securing the valve upon its spindle do not bear hard against the slide, there being a certain amount of lost motion, that is, the spindle travels a distance capable of adjustment from about $\frac{5}{8}$ inch to 1 inch without moving the slide valve. On referring to fig. 62 we see the valve has both ports at each end closed when in its mid position, also that it has neither lap nor lead, and the following explains the working with this arrangement. As the pumps are built

up in pairs the makers design them such that the piston rod of one pump works the slide valve of the *other* one by means of a rocking and a vibrating lever, so that steam may be admitted at each end of the stroke ready for the return one. Thus if the piston of the pump imagined to be behind the one shown in fig. 62 be, say, moving to the right, then it works the small arm connecting to the slide valve of fig. 62 to the *left*, and thus the right hand port (that nearest the gland) becomes opened. And remember whilst the spindle has been moving, taking up a lost motion of from $\frac{5}{16}$ inch to $\frac{1}{2}$ inch, that the piston it has to control has been going towards the right by the steam previously admitted, and we see, therefore, how the valve is controlled properly.

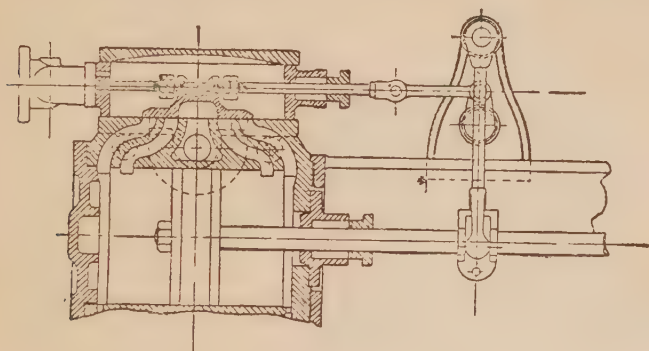


FIG. 62.

The valve in its travel only uncovers the outermost ports and passages to steam inlet, and inner ports, etc., to the exhaust, hence tracing out the actions at, say, the front end of cylinder in fig. 62, we find the *other* pump has moved this slide into a position to admit working steam to the piston, which is then right at the end of its stroke, consequently it begins to return and as the other pump still controls this one's valve this is moved to the left until the outer port is full open. Then the other pump returns due to *its* valve, being controlled by this pump we are now considering, and so the slide of fig. 62 starts to return, but only after the lost motion has been taken up. Thus the valve is left standing for a short time whilst steam is flowing in on one side of piston and the exhaust is open through the *inner* passage and port on its other side. Then the valve moves to the right and closes the steam admission and as soon as the piston in its travel to the left covers the inner passage the exhaust is closed also, and what steam is left at the end of the cylinder is compressed, and so we have the benefits of cushioning obtained from a slide valve that has no steam lap.

By the time the piston reaches the extreme left hand end of its stroke, the *other* pump has moved sufficiently to shift our slide into the position ready to admit working steam at that left end, and so the operations keep on repeating. This seems rather an involved description and to some possibly the following will appeal more :—Calling the pair of pumps, see fig. 63, the front and back ones, when the front steam piston is at the left hand end of stroke it has the back valve right over to the left, with steam port full open and driving back piston to the left hand. This drags along the front valve to the right and so opens the left end of front steam piston, which then moves and pulls the back valve to the right, cutting off the back cylinder's steam

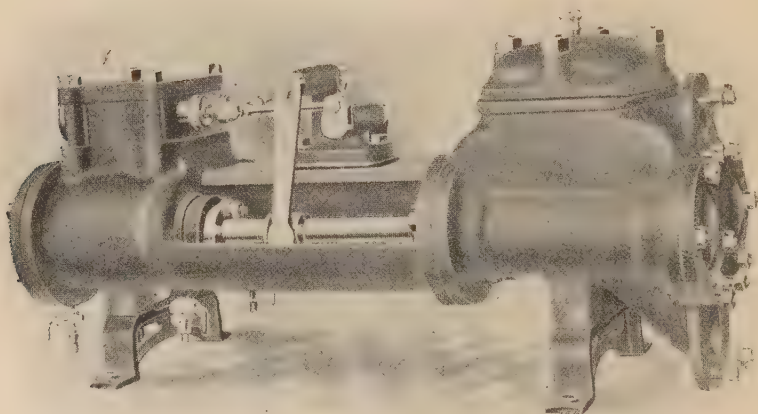


FIG. 63.

and opening up that for its return stroke and when this return stroke has been partly accomplished the front valve is moved to the left, cutting off the front cylinder's steam and opening up that for *its* return stroke. Notice that the back piston and front valve move in the opposite direction, but the front piston and back valve travel together.

Referring to setting the slide valves and the lost motion, remember that if too much of this be given, the piston may foul the ends of cylinder, because the valve will be left standing too long uncontrolled, and, on the other hand, if the lost motion be too little the stroke of pump becomes reduced. To set the valves the following is the procedure as advised by the makers, Messrs. The Worthington Pump Co. :—

Place one piston exactly in the centre of its stroke and the valve of the *opposite* cylinder should then be in its mid position and covering *all* the ports. If the valve rod nuts are then equal distances from the slide valve, *i.e.*, the clearance is the same both ends of valve, the setting is correct, but if this is not the case, then hold the spindle firmly and adjust the nuts and lock nuts until their clearances *are* equal.

Turning now to steering engines we have not the space to do more than describe the principles upon which these operate and point out the functions of the valves, so fig. 64 must be considered as merely

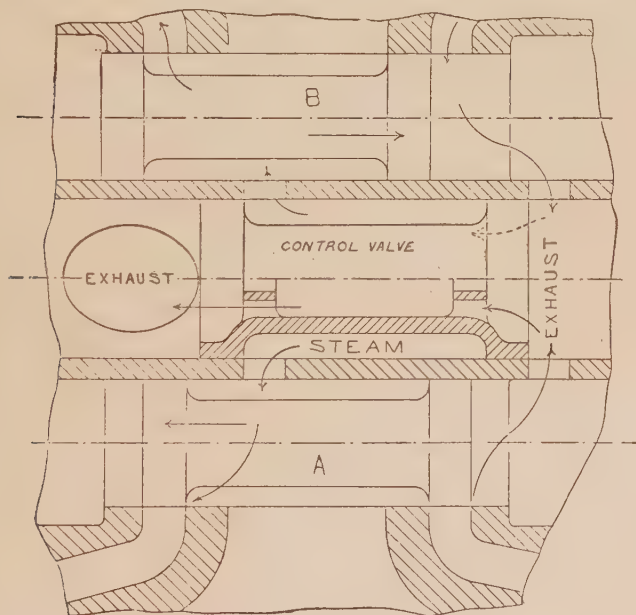


FIG. 64.

diagrammatic and not indicative of any particular maker's design. It shows a set of three slide valves of the piston type, the middle one being the control valve, so termed because whilst it has the steam always on the one side, which may be either inside or outside, yet by movement under the control of the steersman it can admit the working steam over either the inside *or* the outside edges of the main engine valves, and thus "controls" the direction the steering gear shall run in.

To make this most important point as clear as possible refer to fig. 65, which shows a simple cylinder and elementary type slide valve.

Consider the parts moving as indicated and as steam is admitted over the outer edges of slide, the line of throw is ahead of the crank by 90° . Now suppose the engine to have stopped in the position shown in fig. 65, and by some means we are enabled to admit steam into what has hitherto been the exhaust, that is, we send the steam in over the *inner* edges of the valve. It is very evident then that the piston will be driven backwards or in an opposite direction to that at first and consequently the engine *has reversed*, and note also that the one sheave remaining in its original position still answers all right, because, as explained in the early part of this book, when steam is on the inside the

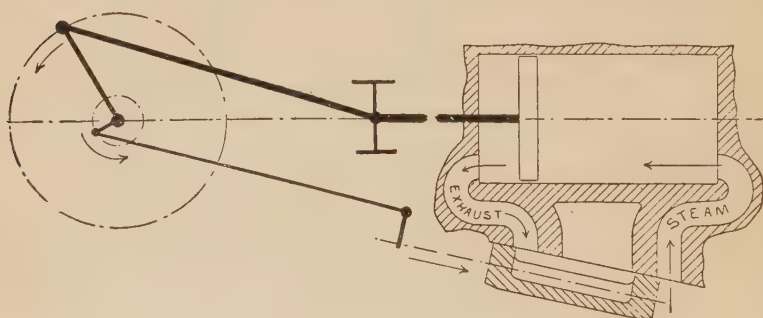


FIG. 65.

sheave's position is *opposite* that for outside steam, and so we now require it to be at 90° *following* the crank which, seeing the engine is running oppositely, is its old placing. Readers are advised to very carefully study this figure, as it lays the groundwork for a thorough understanding of the actions of steering engine valves and explains how it is possible that these gears can reverse without having a special reversing motion, like a "Stephenson" one, and still keep a single fixed eccentric.

We can now consider fig. 64 again, which shows the means we have for alternately admitting steam over either the outer or inner edges of the engine valves *A* and *B*. As drawn, the control valve having *its steam* always supplied over the same edges (here on the inner ones) has the ports leading to the inside of the engine valves open, hence the steering engine runs in such a direction as will have the sheaves following their respective cranks. On stopping the engine what is called the "hunting" gear returns the control valve to its mid position, that is, moves it to the right hand in fig. 64 and so shuts both the ports. Then, if we want the engine to bring the helm over in the opposite direction we, through

the steering wheels, rods, etc., move the control valve out from its central position, only now further still to the right, and thus open the other ports to steam, and which ports acted as the exhaust for the previous direction of rotation. Steam is now sent over the outer edges of valves *A* and *B*, and in consequence the engine runs such that the sheave's line of throw leads the respective crank, thus the steering engine is reversed. There is nothing else to say with regard to the *actions* of these valves, but we may mention that some makers design flat valves instead of piston ones, so do not let the reader think the circular type of slide is essential for steering engines. In a work such as this we cannot describe all details of engine gear, and so those wishful of having the "hunting" mechanism explained should consult some of the well-known books dealing with marine engines and auxiliaries.

CHAPTER XII

THIS deals with some miscellaneous points in connection with slide valve work, and the careful attention of all going in for a Board of Trade certificate is directed to the following.

REQUIRED TO FIND THE TRAVEL OF VALVE.—Add together the following :—Top steam lap + top port opening to steam + bottom port opening to steam + bottom steam lap ; the sum equals valve travel. Or, black the spindle where it is clear of the guiding bracket below steam chest, fix a wire trammel to bear against this blackened part and turn engines for at least one complete revolution, and the point of the trammel will scrape away the black and the clean space on spindle can be measured. Or, subtract the distance between edge of shaft hole on eccentric and the edge of eccentric itself at thinnest or narrowest part, from the distance between the same edges only at its widest part, and the result gives full travel of valve. Or, multiply the line of throw of the sheave by 2 and the product is the valve travel.

TO FIND THE POINT OF CUT OFF OF STEAM.—If from the engine itself, the following is correct :—Place crank, etc., on its top centre by the method explained on page 29. Then, having steam chest cover off, observe the valve has top port open for lead, make a mark on guide and shoe, and then turn the engine towards the bottom. Valve goes down at first and then rises, and as it comes slowly up place a thin piece of paper on its top edge, and the *instant* this is nipped in the port stop turning and see whereabouts the shoe and its mark are. Measure the distance between this and the mark on guide and that is the inches of stroke done up to point of cut off. If it is the bottom port required, then we had better employ valve laths in a similar manner to that explained on page 116, in connection with inside steam admission.

Or, the formula $\left(\frac{2 \text{ lap} + \text{lead}}{\text{travel}}\right)^2 \times \text{stroke} = \text{inches of stroke after cut off}$, and so these subtracted from the whole stroke will give the distance *up to* cut off.

TO FIND LENGTH OF A NEW ECCENTRIC ROD, THERE BEING NO OLD ROD IN PLACE.—Place the valve in its mid position, and take the distance from centre of tumbling block at end of spindle down to centre of the shaft, and from this subtract half the diameter of the eccentric plus the thickness of top half of the strap. The remainder is the length of rod between centre of brass at its forked end to the flat underside of its foot.

TO PUT A SLIDE VALVE IN ITS MID POSITION.—Proceed as stated for finding the valve travel by blackening the spindle. Taking care that the trammel is fixed so that it cannot move, bisect the distance it has cleaned upon the spindle, and then bring this point of bisection under the scrieving end of trammel, when valve will be in its mid position.

TO MINIMISE FACE FRICTION OF HEAVY SLIDES.—Cut a series of grooves (similar to oil grooves for a bearing) along each face and down each side. The moisture in the steam collects in these and serves as a lubricant, thus easing friction, as, remember, it is the best policy to get an engine to run with as little oil for her internal lubrication as is safely possible, on account of this oil getting into the boilers and causing at times very serious trouble therein. We do not advise total exclusion of all oil, as possibly not one engine in 10,000 could run for two hours like this, but doing a simple thing like cutting the above grooves is a decided help.

REQUIRED THE LENGTH OF A NEW SPINDLE.—Place engine on its top centre, and by means of blocks and shores get the valve against the cylinder face, and having the top port open for the requisite lead. It is presumed the link motion is all up and not damaged, and so bring the headway rod into what *would be* in line with spindle. Then procure a long rod either of iron or wood, and drop this down through the spindle hole in valve and also through the stuffing box until it comes to the exact centre of the reversing link. Then scribe a mark on this rod exactly where the bottom and top of valve come to, after which the rod can be withdrawn from the chest, and allowance made for the thickness of two nuts, and sufficient spindle to allow either the split pin or split collar to go through above them.

WHAT TO DO IF THE H.P. VALVE BREAKS.—Take out all the broken pieces, and if no spare valve we must work without this cylinder thus:—Leave the spindle in the stuffing box, but if bent at all take down the link motion so that spindle does not move. Reduce the pressure of steam in boilers *before* starting again, and as the steam flows in both top and bottom of piston as well as through the exhaust port on cylinder face the piston is in equilibrium and the I.P. cylinder gets its

supply of steam all right. The amount of reduction in the pressure must be such as leaves it only a trifle higher than what the I.P. has hitherto been working with.

WHAT TO DO IF THE I.P. VALVE BREAKS.—Take out the broken pieces and do as above with regard to the spindle. There is now no need to reduce boiler pressure, as the H.P. engine is working all right and only allowing the usual amount of steam to come into the system, and as the exhaust from the H.P. coming into the I.P.'s chest finds it can get into both top and bottom of the I.P. piston as well as away to the L.P. cylinder, the pressure in the I.P. chest may actually be a little lower than it was before.

WHAT TO DO IF THE L.P. VALVE BREAKS.—Take out the broken pieces and do as before in regard to the spindle. As the steam does no work in this L.P. cylinder now, and so would go to the condenser hotter than before, it may be of benefit to alter the I.P. valve a trifle in order to make it cut off sharper and so exhaust its steam at a lower pressure and temperature into the L.P. Thus the vacuum would not be so seriously impaired. One has to remember, though, in any question of an engine breakdown, that they must be almost wholly governed in what they are to do by the design of the engine itself, and the conditions pertaining at the time of the accident, and all we can do in these pages is to lay before one what should or might be effected, and leave it to be seen whether such can actually be carried out when the smash occurs.

SUPPOSE THE SLIDE TO BE LOOSE ON ITS SPINDLE, HOW WOULD THIS AFFECT THE STEAM DISTRIBUTION?—By the slide being slack or loose we mean that the nuts or collar upon its spindle have worn, and there is a certain amount of lost motion, that is, when the spindle attempts to say, pull down the valve, that it moves for the amount of this wear *before* the valve follows it and thus certain of the actions are later. Before we can answer definitely which will be so affected, we must know whether the wear is at the collar or the nuts. Let us suppose it to be the former. Say the collar has worn $\frac{1}{16}$ inch, then everything due to the valve being pushed *up* will be later, but all actions due to its coming down will be the same as before, because the nuts are just the same. Therefore we should find cut off on top later, on bottom correct; lead at top correct, but late on bottom; exhaust opening later on top and closing correctly, as then the valve is coming down again. Had it been the nuts which had worn, then the opposite of the foregoing would be the result, as it then would be, to a certain extent, the equivalent of the valve being *high*, whereas a worn collar

makes it too *low*, but remember that with a loose valve the travel is not so great. In practice, though it is certain that any looseness such as we are writing of is produced by wear, both of the collar and the nuts, in which case, if the amount be considered as equally divided between both, we have for $\frac{1}{16}$ inch slack, $\frac{1}{32}$ inch the valve is out of setting considering its mid position, and so when it is being pulled down all actions will be upset by this $\frac{1}{32}$ inch, and, similarly, when being pushed up everything is upset by $\frac{1}{32}$ inch also.

REQUIRED TO KNOW WHAT CLASS OF ENGINE BENEFITS MOST BY HAVING STEAM LAP.—A marine engine receives most benefit from its slide valves having steam lap, as by so fitting them we are enabled to use steam of higher pressures and so obtain permanent economy in coal consumption, this in turn leading to a saving in bunker space, and if the owners of the vessel like they can have this space thrown into that in which cargo is carried, and so freight may be earned. There is thus a twofold gain in the case of a marine engine.

REQUIRED TO KNOW WHETHER A SLIDE VALVE IS SET CORRECTLY AND IS WORKING IN A SATISFACTORY MANNER WITHOUT CHECKING OVER FROM THE ENGINE ITSELF.—It might be thought that if the engine was revolving in a smooth and regular way that the valves were all set quite correct, and, further, if the coal consumption was reasonable we might be inclined to say there was nothing wrong anywhere, and yet the slide valves might not be as efficient as possible or yet set correctly. We must take a set of indicator cards from each of the cylinders, and see that the instrument itself is coupled up properly and works easily, is clean, and warmed through before taking the diagrams. Then by a careful inspection of these we can see how the setting of the valve is. In the following we purposely write out at full length the various descriptions of possible faults, as everyone having to deal with indicator diagrams should be able to speak of the different lines, points, and actions, instead of only pointing them out on a card when one is shown them:—

To commence, then, we may assume our reader to be acquainted with a card, and to know whereabouts such points as cut off, exhaust opening and closing, lead line, etc., are, because this book does not profess to teach all about an engine's adjuncts, but merely slide valves and what is immediately in connection with them for the average engineer. Should our card show us the exhaust opening at a point earlier than about $\frac{1}{3}$ of the total length of diagram, we say that point is *early*. If the exhaust closes at a point earlier than about $\frac{1}{3}$ of the stroke we say that is *early* also. Similarly, the opening to exhaust

should not occur appreciably later than about the $\frac{1}{9}$, and if it does, we should say the exhaust opened *late*. Then for the cut off we cannot say with any great degree of certainty where this should be, because we may not know what work that cylinder has to perform, and it may be arranged for either a late, medium, or early cut off. We can, however, get a very fine idea as to the lead, and if we see this line sloping *well away from* the diagram then we know the lead is too great, but should it slope *in towards* the card itself then we know there is an absence of lead, as by rights the line itself should be practically vertical. Having from the indicator card determined what the errors in setting of the valve are, we can think of what has to be done to put matters right, and here we would advise a most careful studying of the earlier pages of this book.

Further than only noting errors in setting, the indicator card tells us a great deal as to the condition of the faces, both of cylinder and slide valve. Thus if we see the expansion curve on the diagram rising outwards, instead of constantly dropping, we know at once that the steam is flowing into the cylinder even after the point of cut off, and to do this there must be a leak existing between the two faces of cylinder and slide. Then if we see the admission steam line on the card drooping down a great deal, and we know the diagram is the H.P. one of, say, a triple-expansion engine, we at once conclude the steam is being throttled, that is, the area of port open for its admission is so small and is so rapidly being reduced by the moving slide valve as to baffle the entering steam sufficiently to make it lose pressure, and the remedy here will be to make the ports larger, though in practice this is almost an impossibility.

IT HAS BEEN STATED IN AN EARLIER PART OF THIS BOOK THAT RELIEF RINGS ARE FITTED TO REDUCE VALVE FRICTION. DO YOU KNOW ANYTHING ELSE OF A SOMEWHAT SIMILAR NATURE FITTED FOR THE SAME PURPOSE?—Yes, sometimes such a valve as a large double ported one is designed and made with a passage through its back and communicating with a corresponding enlargement of the L.P. steam chest back, which portion is in direct connection to the condenser by having the large eduction pipe fitted to it. On the back of the slide there is fitted a relief ring in much the same style as those we have already described, and provided this is kept steam tight, as it always should be, the above design makes a very nice method of ensuring a free exhaust, and at the same time relieving a large amount of the valve's surface from the effect of live steam.

Whilst on this subject of a relief to the back of a valve we would

draw attention to the fact that very often slides have to be kept up to the cylinder face by artificial means, such as springs, guides, etc., and which can exert a pressure upon the back of the valve, and lest the reader should think this is entirely at variance with what we wrote earlier about taking off some of the pressure, let us explain what is really meant.

When a valve is small the effort to overcome friction is not, of course, so great, given the steam pressure equal in both cases, to one that is larger, and hence a relief ring is not so necessary, in fact, very few smallish slides have these fitted, and so the steam pressure acts on the whole back. Now at times the compression of the steam in the cylinder may, through bad setting of the valve, become even higher than the pressure in the chest, and so there is a decided tendency to force that end of the valve away from the cylinder face and thus allow escape of the cushioning steam, to say nothing of the tendency to bend the valve spindle if the slide actually does leave the other face, and so to prevent this we often see valves with springs, etc., fitted to keep them up to the face. Of course, all such adds to the friction, but as a possible leakage is of vastly more importance than a little increase in friction we choose the lesser evil. With big valves such means are usually not so necessary, as the area open to compression is relatively less in comparison to the whole area of back for the live steam to act upon, and this "easing" tendency does not cause much trouble.

One will also find guiding strips fitted on the cylinder face, down each side of where the edges of the slide come to, and it is inside these that the valve works, being when cold a rather slack fit between these guide bars, whose function is to ensure the valve properly working in a *straight* line without entirely relying upon its spindle to do this, and thus we are led on to another detail, viz., the fitting of the external guide bracket below the steam chest and stuffing box.

If the engine were a non-reversible one, and the eccentric rod was enormously long in comparison to the line of throw of eccentric, then possibly a good long stuffing box, with a gland well supported on four or even six gland studs (instead of the usual two), would doubtless guide the spindle correctly and without undue wear and tear. But directly an engine has to reverse, every time the link is pulled over a bending stress is set up on the spindle, and which, when the gear is in such a position as shown by *B* or *C*, plate I., becomes very severe, and would quickly result in the spindle wearing the gland and neck bush oval, and as soon as even the least wear occurred it would rapidly become worse. Further, the spindle might break, because we never

know but what it has a flaw, that so long as steady stress is brought upon it lasts out, but when the inclined reversing link begins to pull or push at the end of the spindle the link goes over in more or less a series of jerks, at every one of which a little more wear is brought about. To minimise as much as possible the trouble, a second guide is fitted as low down towards the tumbling block as the travel of the same will permit, and so, instead of the spindle jamming in the gland and neck bush directly the act of reversing tends to bend it over one way or the other, this external guide supports the lower end by virtue of its brasses embracing that part of the spindle, hence the engine reverses easier and with more safety.

REQUIRED THE METHOD OF SETTING A SLIDE WHEN THE PORT ITSELF CANNOT BE SEEN, SUCH AS FOR THE LOWER PORT OF A MARINE CYLINDER (STEAM OUTSIDE) OR EITHER PORT IF THE STEAM BE INSIDE.—Resort must be had now to valve “laths,” and we proceed to show how these must be made and placed. Our remarks are applying only to inside steam admission, but it must be remembered the principles underlying are the same in any case where such laths have to be employed. Having provided two parallel laths a good deal *longer* than what we know the valve to be over its outer edges, we see that at least one of the lath edges on each is straight. Next, before removing the piston slide from its spindle, scribe a line all round the chamber it works in, where the top edge of valve happens to be at, and then the valve can be taken out. Next lay a lath down the chamber, and mark plainly upon its straight edge where the steam and exhaust ports come, and also the position of that special line we scribed round the chamber. Mark also upon the other lath exactly where the steam and exhaust edges of the valve come, and afterwards we can replace the slide upon its spindle and tighten up the nuts.

Now at some convenient spot outside the steam chest erect the two laths, the valve one coming from the valve or its spindle, and being firmly connected thereto by any means suitable for the particular engine. Against this lath erect the cylinder face one in such a manner that the edges of the two laths are practically touching each other, and also, and what is of the utmost importance, see that they are put up so that the top edge of valve comes *exactly* to the mark scribed to show its position on the cylinder face one, then fix in that place. If now the engines are turned round we shall be able to follow the openings and closings of the ports by the valve merely from watching the valve lath move against the cylinder face one, and so all questions of setting or adjustment can be carried out by either moving the eccentric sheave

to or fro upon the shaft, or putting in or taking out liners as usual, and when we see the desired change brought about from inspection of these laths we know the same has occurred with the actual valve.

Having brought this work practically to a finish, we will actually conclude by giving a little advice to all those preparing to study the slide valve and the gear used to drive it, especially intending candidates for Board of Trade certificates. When a young engineer is serving his apprenticeship in the "shops" he should pay particular heed to any and every part relating to valves and valve gearing that comes his way for machining, fitting or erecting, and especially this latter.

He should take careful note of the various designs he comes across and be particularly observant as to the sizes of everything, because, when he comes to sit his exam. for chief engineer he may very simply be set as a drawing, a sheave strap and rod, or a reversing link, or perhaps a complete link motion. We would advise every young engineer to provide himself with a sketch book and on an opportunity offering, let him make sketches of whatever he comes across, not merely in connection with slide valves, but with all the other parts of the engines, and be sure to mark in all, or at least as many as can be managed, sizes upon these sketches.

Then if engaged upon the erection of, say, a set of marine engines, let him watch very carefully how the valves are set and adjusted, and though in many cases the methods employed in the shops are different to what may have to be done when afloat, yet much valuable knowledge can be gleaned by an intelligent observation of what goes on around one.

On his going to sea, let him note how all the link motion, sheaves, etc., are running, provided his chief has got things into ship-shape order, and our young engineer will then be able to see for himself what he has to maintain during his watch. Should there be trouble with the gear, then let him take particular note as to what indications it gives of trouble, how his superior engineers set about remedying matters and how it runs after they have worked at it, but by no means worry them unduly by asking them a lot of questions, the bulk of which his own eyes and common sense should have answered for him. The majority of seconds and chiefs are only too glad to help a junior engineer when they see he honestly wishes to learn and so improve his position, but a man who simply asks questions for asking's sake is only a nuisance and an irritant on board a ship.

Having carefully read through the pages of this book and understood the different statements as far as one can—and the writer feels tempted

to hope that this may not be difficult to carry out fully—careful watching of the gear when at work and noting the constructions and settings when it is being overhauled will, we trust, give one a very good insight into the behaviour, etc., of slide valves and valve gearing, and with this hope we place the book in the hands of the engineering public, especially those whose duties lie at sea.

INDEX

INDEX

	PAGE		PAGE
Ability to reverse steam steering gear	108	Bars for Meyer expansion valve	77-78
Action of valves, Weir's pump	100	Best condition of engines running	87
Actions with elementary valve	5-8	Block, sliding, Hackworth gear	71
Actions with valve having steam lap	10-15	Bolts, coupling, undue stress on	82
Actions with Hackworth gear	71	Breaking of H.P. valve, what to do	111
Actions with Marshall's gear	72-73	Breaking of I.P. valve, what to do	112
Actions with Joy's gear	75	Breaking of L.P. valve, what to do	112
Additional lap for sharper cut off	68	By-pass ports, Weir's pump	100
Adjusting length of stroke, Weir's pump	103-104	Caution when altering cut offs in Meyer expansion valve	80
Adjusting horse power, gain by	89	Cavity, exhaust in valve	3
Adjusting horse power between cylinders	82	"Check" marks on shaft and sheave	31
Adjusting and altering leads	35	Class of engine to benefit by steam lap	16-17, 113
Advantages with single sheave gears	74	Closing in an engine, effect of	88
Advantages with piston valves	44-47	Coal consumption decreased by balancing horse power	89
Alternative to simple relief ring only	114	Completion of stroke by expansion	6
Altering cut offs, Meyer gear, caution when	80	Compression in cylinder	3-14
Alteration in leads, table of	39	Constant lead when linking up with single sheave gears	70
Angle of advance, definition of	13	Connecting rod angularity, effect of	65
Angular distance between lines of throw	27	Coupling bolts, undue stress on	82
Angular advance, extra for lead	15	Crank on top centre, putting	29
Angular movement of sheave in comparison to linear movement of valve	28	"Crossed" eccentric rods	55
Angularity of connecting rod, effect of	65	Curvature of reversing link	55
Angle of bars in piston valve chamber	45	Cushioning or compression,	3-14
Areas of top and bottom, "pistons" of valve	46	Cut off sharpened by linking up	57
Arrow to show direction crank turns in, keyway diagram	27	Cut off with a donkey crosshead	64-65
Ascertaining friction of slide valve	91	Cut off sharper, additional lap for	68
Ascertaining lead by means of wedge	31	Cut off, how effected by expansion valve	79
Assistant cylinder, Joy's	96	Cut off, index for	80
Auxiliary valve, Weir's pump	101	Cut off, finding point of	110
Balance weight for loose eccentric	76	Cutting keyways in new shaft	30
Balance cylinder and piston	95	Cylinder face, for double ported valve	40
Balancing horse power between cylinders	82	Cylinder ports and passages for piston valve	46
Bar, middle, with double port slide valve	42	Cylinder, balance	95
Bars in piston valve chamber, inclination of	45	Definition of slide valve	2
		Definition of "face" of slide	3
		Definition of steam lap	8
		Definition of angle of advance	13
		Definition of "eccentric" or "sheave"	22
		Design of piston slide valve	44
		Description of reversing link	55
		Description of Plate I.	60

	PAGE		PAGE
Diagram, keyway	25-27	Expansion valve, Meyer's	77
Diagram, lap finding	65-68	Expansion valve, use of	77
Difference in opening and closing exhaust, with exhaust lead, or lap	20	Expansion valve, position of sheave for	78
Difference in sheaves' position if steam inside valve	27	Expansion valve, how it effects the cut off	79
Difference in area of top and bottom pistons of valve	46	Expansion valves, where fitted	80
Difference in cut offs top and bottom	62	Extra angular advance for "lead"	15
Distance between points of eccentric rod attachment on reversing link	55	Face of slide, definition of	3
Division of total horse power in triple compound	83, 87-88	Face of valve, depth of, with exhaust lead	20
Donkey crosshead	62-63	Face of valve, depth of, with exhaust lap	20
Donkey crosshead, cut offs with	64-65	Face cylinder for double ported valve	40
Double ported slide valve	41	Faults with single sheave gears	73
Drawing keyway diagram, care when	27	Faults with relief rings	94
Drawing lap finding diagram	68	Finding lead by means of wedge	31
Drawing the valve	22	Finding point of cut off	110
Easing valve friction by relief rings	93	Finding travel of valve	110
"Eccentric," meaning of	22	Finding length of new spindle	111
Eccentric or sheave	22	Finding length of new eccentric rod	111
Eccentric, correct placing	23	Finding lead, etc., when port cannot be seen	116
Eccentric rods, open and crossed	60	Forked ended eccentric rods, revers- ing with, etc.,	53
Eccentric, single, Hackworth	70	Free exhaust, need for	2
Eccentric, single, Marshall	72	Friction of valve, trying to ascertain	91
Eccentric, none, Joy's	74	Functions of slide valve	2
Eccentric, loose	75-76	Gab lever motion	76
Eccentric, loose, following crank	77	Gain by adjusting horse powers	89
Eccentric rods, new, finding length of	111	Gear, Stephenson's, faults with	70
Economy of high pressure steam	49	Gears, patent valve	70
Edges, steam and exhaust of valve	4	Gear, Hackworth's	70-71
Effect on exhaust with minus lap	17	Gear, Marshall's	72
Effect of exhaust lap	20	Gear, Joy's	74
Effect on leads by linking up	60	Great size of double ported valve	41
Effect of angularity of connecting rod	65	Great variation in loads by very sharp cut offs	52
Effect of opening out engine	88	Hackworth's valve motion	70
Effect of closing in engine	88	Heat, latent	50
Effect of opening out or closing in an engine	87-89	Heat, sensible	50
Effect of independent linking up	89	Heat, to evaporate 1 lb. water	50
Effect of valve loose on spindle	112	High pressure steam economy with	49
Elementary type of valve	5	High pressure valve, Weir's pump	102
Engines suited for steam lap	16, 114	High pressure valve breaking, what to do	111
Engine running, best condition of	87	Horse power, balancing,	82
Equality of cut offs with donkey crosshead	64-65	Horse power, balancing gain by	89
Equality of cut offs, endeavouring to produce	67	Horse power, balancing table of effects	90
Exhaust, need for free	2	Importance of slide valve	1
Exhaust cavity in valve	3	Inclination of bars in a piston valve chamber	45
Exhaust lead	17	Increase in depth of valve face by exhaust lap	20
Exhaust lap	20		
Exhaust straight through back of valve	114		
Expansion of steam not sole cause of economy	51		

	PAGE		PAGE
Increase in speed by balancing horse power	89	Marshall's valve gear	72
Independent linking up,	80-81, 89	Meaning of "eccentric"	22
Index for cut offs, expansion gear	80	Meaning of phrase "put sheave forward $\frac{1}{8}$ "	28
Inside steam admission	34	Meyer's expansion valve	77
Iron template	25-27	Method of setting valve when port cannot be seen	116
I.P. valve breaking, what to do	112	*Middle bar on cylinder face, for double ported valve	41
Joy's valve motion	74	Minus lap	17
Joy's assistant cylinder	96	Miscellaneous questions	110
Keyway diagram	24, 25-27	Need for free exhaust	2
Keyway diagram, care when drawing	27	New shaft, cutting keyways in	30
Keyways, cutting in new shaft	30	New spindle, finding length of	111
Lap, steam, definition of	8	New eccentric rod, finding length of	111
Lap finding diagram	65-68	"Open" eccentric rods	55
Lap extra, for sharper cut off	68	Opening out an engine	88
Latent heat	50	Opposite actions, steam inside	35
Laths, valve and cylinder,	116	Patent valve gears	70
Lead	16	Path of eccentric sheave	22
Lead, what governs its amount	16	Piston type of slide valve	43-45
Lead, exhaust	17	Piston valve, advantages with	44
Leads, adjusting and altering	36	Pistons, difference in area of top and bottom	46
Leads, table of alteration in	39	Piston's balance	95
Leads, setting of, with Tric and double ported valves	48	Plate I., description of	60
Leads, effect on by linking up	60-61	Ports and passages to cylinder with piston valve	47
Leakage of steam with relief rings	94	Position of elementary valve, crank on top centre	5
Leak, steam lost through	94	Position of elementary valve, crank horizontal	6-9
Length of eccentric rods	55	Position of elementary valve, crank on bottom centre	7
Lever, gab	76	Position of crank, when valve with steam lap is middled	11
Line of throw	22	Position of crank, when valve with steam lap has cut off	12
Lines of throw, angular distance between	27	Position of crank, when valve with steam lap has bottom port full open	13
Linear alterations in valve setting	32	Power, total, division in triple compound	83, 87-8
Linking up	54	Pressure and volume of steam, relation between	82-83
Link motion	54-55	Pump, Weir's	97
Link, reversing, radius of	55	Pump, Weir's by-pass ports	100
Link, description of reversing	55	Pump, Weir's views of valves, etc.	99-102
Link, motion actions of	56	Pump, Weir's main and auxiliary valves	101
Link, middled, valve travel with	59	Pump, Weir's adjusting stroke	103-104
Link, swinging, Marshall gear	72	Pump, Worthington's	104
Linked up, reduction of travel when	58	Pump, Worthington's valves to set	107
Linking up, effect on leads	60	Pump, Worthington's lost motion to valve	104
Linking up, independent	80-81	Pump, Worthington's action of valves	105-106
Load, great variation in, by very sharp cut offs	52	Putting crank on top centre	29
Loose eccentric	75-76		
Loose eccentric balance weight for shaft	76		
Loose eccentric stopper for, on shaft	76		
Loose eccentric following crank	77		
Loose valve on spindle, effects with	112		
L.P. valve breaking, what to do	112		
Loss of work from increased back pressure	16		

	PAGE		PAGE
Quantity of heat to evaporate water	50	Slide valve, design of piston type	43 & 45
Quantity of steam lost by leak	94	Slide valve, "Tric" type	47-48
Questions, miscellaneous	110	Slide valve, definition of faces	3
Radius of reversing link	55	Slide valve, steam and exhaust edges of	4
Reduction of face depth, by giving exhaust lead	20	Slide valves, Weir's pump	98-102
Reduction in consumption by balancing power	89	Sliding block, Hackworth gear	71
Reduction in valve travel by linking up	58	Slotting action with reversing link	56
Reference or check marks on shaft and sheave	31	Spindle, valve loose on	112
Relation of crank and valve (elementary type)	5-9	Spindle, new, finding length of	111
Relation of crank and valve (with steam lap)	10-15	Speed, increase in by balancing power	89
Relation between pressure and volume of steam	82-83	Steam steering gear valves	107
Relief rings	93	Steam steering gear ability to reverse	108
Relief rings, faults with	94	Steam pockets, double ported valve	40-41
Reversing with forked end eccentric rods, etc.	53	Steam lost through leak	94
Reversing link, description of	55	Steam lap, definition of	8
Reversing engine with a loose eccentric	76	Steam lap, engines suited for	16
Reversal of steam steering gear	108	Steam on inside of valve	34
Saving by using high pressure steam	51	Steam, high pressure, economy with	49
Saving erroneously attributed to "expansion"	51	Stephenson gear, faults with	70
Scribe marks as reference, on shaft and sheave	31	Stopper on shaft for loose eccentric	76
Section through steam cylinder of Worthington pump	105	Swinging link, Marshall gear	72
Section through Weir's pump	98	Table of laps and their effects	21
Setting Worthington pump valves	107	Table of alteration in leads	39
Sensible heat	49	Table of alterations by sheave and by liners	33
Setting a slide valve	30	Table of effect on leads by linking up	61
Setting slide valve when ports cannot be seen	116	Table of effects produced by balancing horse powers	90
Sharpening cut off by linking up	57	Template for key position	25-27
Sheave, path of	22	Top port open for "lead"	15
Shaft stopper, loose eccentric	76	Top centre, putting crank on to	29
Shaft, new, cutting keyways in	30	Top and bottom cut offs, difference between	62
Sharper cut off, extra lap for	68	Trammel used when placing engine on top centre	29
Sheave, correct placing	23	Travel of elementary valve	10
Simple slide valve	5	Travel of valve with steam lap	12
Single sheave gears, constant lead with	70	Travel, reduction in by linking up	58
Single sheave gears, faults with	73	Travel of valve when link middled	59
Single sheave gears, advantages with	74	Travel of valve, finding	110
Slide valve, importance of	1	Tric valve	48
Slide valve, definition of	2	Triple compound, division of total power	83, 87-88
Slide valve, functions of	2	Trippet pin, loose eccentric motion	76
Slide valve, elementary type	5	Trying to produce equality of cut offs	67
Slide valve, piston type	33, 43-44	Trying to ascertain valve friction	91
Slide valve, double ported	40	Type of valve for steam steering gear	5
		Undue stress on coupling bolts	82
		Upper and lower faces of valve	3
		Valve, slide, importance of	1
		Valve, slide, definition of	2
		Valve, slide, functions of	2
		Valve, face, definition of	3

	PAGE		PAGE
Valve, elementary type	5	Water, heat to evaporate 1 lb. ...	50
Valve, positions in relation to crank	5-15	Wedge, used for finding amount	
Valve travel with elementary type	10	of lead	31
Valve, travel of, with steam lap ...	12	Weir's pump	97
Valve, driving the	22	Weir's pump, sectional view ...	98
Valve setting,	30	Weir's pump, sectional plan and	
Valve setting altered by liners ...	32	front elevation	99
Valve setting when ports cannot be		Weir's pump, action of valves ...	100
seen	116	Weir's pump, auxiliary and main	
Valve, slide, piston type	34, 44-45	valves	101
Valve, slide, double ported	41	Weir's pump, "high pressure"	
Valve, double ported, great size of	41	valve	102
Valve piston, advantages with ...	44	Weir's pump, adjusting stroke	103-104
Valve, Tric pattern	47-48	Weir's pump, by-pass ports ...	100
Valve travel, reduction in by link-		What governs the amount of	
ing up	58	lead	16
Valve motion, Marshall's	72	What happens when a liner is	
Valve motion, Joy's	74	inserted	33
Valve motion, Hackworth's	70	What to do if H. P. valve breaks ...	111
Valve gears, patent	70	What to do if I. P. valve breaks ...	112
Valve motion, gab lever	76	What to do if L. P. valve breaks ...	112
Valve, Meyer's expansion	77	Why linked up gear cuts steam off	
Valve friction	91	sharper	57
Valve friction, relieving	93	Why travel is less when linked	
Valve, auxiliary, Weir's pump ...	101	up	58
Valve, main, Weir's pump... ..	99 & 101	Why lead remains constant with	
Valve, actions of, Weir's pump ...	100	single sheave gear	73
Valves of steam steering gear ...	107	Worthington pump... ..	104
Valve control	107	Worthington pump, lost motion of	
Valve travel, finding	110	valve	104
Valve loose on spindle	112	Worthington pump, section of	
Valve with exhaust through back	114	steam cylinder	105
Valve laths	116	Worthington pump, action of	
Variation in loads by very sharp		valves, etc.	105-106
cut offs	52	Worthington pump, to set the	
Volume and pressure of steam, re-		valves	107
lation between	82-83		

